A STUDY OF THE FAILURE OF JOINTS IN COMPOSITE MATERIAL FUEL CELLS DUE TO HYDRAULIC RAM LOADING

Henry Speer Ezzard

NAVAL POSTGRADUATE SCHOOL

Monterey, California

THESIS

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IN COMPOSITE MATERIAL FUEL CELLS DUE TO
HYDRAULIC RAM LOADING

by

Henry Speer Ezzard, Jr.

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A Study of the Failure of Joints in Composite Material Fuel Cells Due to Hydraulic Ram Loading

by

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Ensign, United States Navy
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ABSTRACT

The objectives of this research were to show the relative importance of the transverse shearing forces, the bending moments, and the tensile forces produced by hydraulic ram loading on military aircraft fuel tank joint designs for composite materials, and to present fuel tank test section designs. With the use of a finite element analysis, it was shown that the transverse shearing force may be a major cause of attachment failure of composites, primarily by an unzipping or pull out mode of failure. It was also shown that failure criteria for transverse shearing stresses in composites are lacking. By comparing several specific aircraft design concepts, designs for a wing fuel test tank and a fuselage fuel test tank were selected. Simplicity, similarity to actual aircraft fuel tanks, and uniformity between experiments were primary considerations in the selection.

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LIST OF SYMBOLS

a	-	t/2
b	-	base dimension
D	-	diameter
E	-	modulus of elasticity
F	-	force
G	-	shear modulus
I	-	moment of inertia
M	-	bending moment per unit length
N	-	tensile force per unit length
Q	-	transverse shearing force per unit length
R	-	radius
S	-	stress
t	-	thickness
Wt	-	density
x	-	axis along spar
y	-	axis perpendicular to spar, load application axis
z	-	axis through the thickness
σ	-	normal stress
τ_{yz}	-	transverse shear stress
ν	-	Poisson's ratio

SUBSCRIPTS

1	-	longitudinal axis
2	-	transverse axis
11	-	y-axis
22	-	z-axis
12	-	45^{o} between 11 and 22 directions
xx	-	x-axis
yy	-	y-axis

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I. INTRODUCTION

Laminated fibrous composite materials, such as graphite-epoxy and boron-epoxy composites, are currently being considered for large scale use in military aerospace vehicles because of their high strength-to-weight ratios. For example, a version of the F-18 with graphite-epoxy wing skins is in the design stage. The analytical methods currently being used for the design and analysis of these inhomogeneous, anisotropic composite skins are valid for the internal plane stress (membrane) conditions caused by the normal aircraft loads. The problem of interest here is not that of the stresses caused by the flight loads, but is rather the determination of the damage to a composite fuel tank wall due to a penetrating projectile, such as might occur in a combat situation.

Projectiles which penetrate fluid containing tanks can cause damage much more severe than that incurred by penetration through an empty tank. Hydraulic ram loading is the phrase used to describe the destructive fluid loading on the tank walls. General internal loading conditions in the tank walls of tensile forces, bending moments, and transverse shearing forces due to the hydraulic ram pressure

have been identified in earlier studies. Hydraulic ram tests on composite plates at the Naval Postgraduate School [Ref. 8] and hydraulic ram tests on aluminum plates by Ultrasystems, Inc. [Ref. 9] have led to the identification of the above loads through identification of the plate failure modes. The bending moments and transverse shearing forces, of little consequence in normal aircraft operations, present a formidable problem to aircraft fuel tank integrity when the tank has suffered a ballistic impact. The analytical methods for analyzing and designing composite skins with large bending moments and transverse shearing forces are either nonexistent or of questionable validity.

The objective of this study is to investigate the effect of hydraulic ram loading on the fuel tank design concepts currently being considered for composites. In particular, the area around the attachments of the composite skin to the supporting members, such as spars and ribs, will be examined in detail because these are the regions of highest bending and shear stresses. Tests performed at the Naval Postgraduate School [Ref. 8], the Northrop Corporation Aircraft Division [Ref. 5], and at Ultrasystems, Inc. [Ref. 9], have given indication that the attachment regions are the most critical failure regions of the tank when under hydraulic ram loading.

In order to study the attachment problem, a definition of joint failure possibilities for the composites is presented. A typical fastener joint is selected for analysis. A finite element analysis of the joint is conducted to investigate the effect of unit bending moments/unit length, unit transverse shearing forces/unit length, and unit tensile forces/unit length, on the stress field in the skin around the fastener. The finite element program, SAP IV [Ref. 6], was used to create both a plate bending model and a local two-dimensional thickness model for an isotropic material, to determine the regions of high stress for conventional metal fuel tank walls. A comparison of these results with the results from similar analyses for composite walls (to be conducted in another study), in conjunction with a failure analysis, will reveal whether or not the attachment problem is more severe for composite fuel tanks than it is for metal tanks. Several attachment designs to reduce the hydraulic ram damage to the fuel tank are also presented.

II. FRACTURE AND FAILURE OF THE COMPOSITE DUE TO HYDRAULIC RAM LOADING

The different modes of fracture in a composite fuel tank, due to the impact and penetration of a projectile through the fluid-containing tank, have been identified with the use of a ballistic range. Ballistic tests have been performed at both the Northrop Corporation Aircraft Division [Ref. 5], and the Naval Postgraduate School [Ref. 8], and have provided results on the types of fracture due to the hydraulic ram loading on two specific wall designs. In general, the study of composite fracture is extremely difficult due to the many parameters that affect it, such as the lay up of the laminate, the material properties of the laminate, the thickness of the laminate, the depth of the fluid, the size of the panel being fired into, and other parameters.

At present, five main types of fracture of the composite fuel tank wall have been identified. Although there is always some form of hole in the wall due to the penetration of the projectile, different crack types may occur. First, there is the possibility of radial cracks extending out

from the hole, as illustrated in Figure II.1. Second, there is the possibility of a lengthwise crack with extensive delamination of the material around the hole, as seen in Figure II.2.

The remaining modes of fracture do not occur around the hole, are significantly influenced by the hydraulic ram loading, and indicate a more serious loss of structural integrity than the first two modes of fracture. The third fracture type has long circular cracks near the edges or attachments of the plate, where a bonded attachment technique was used. It is illustrated in Figure II.3. The fracture in the fourth case is very similar to that of the third case. This type also has long cracks along the edges of the plate, where the attachment to the supporting members is made, but the attachment technique used here was one of clamped edges, and the fracture has almost led to a breakout of the entire plate, as illustrated in Figure II.4. The main difference between the third and fourth types of fracture is a difference in the method of attachment of the plate. In the third case, the attachment was a built up, interlaminarly bonded joint. The attachment in the fourth case was a clamped edge.

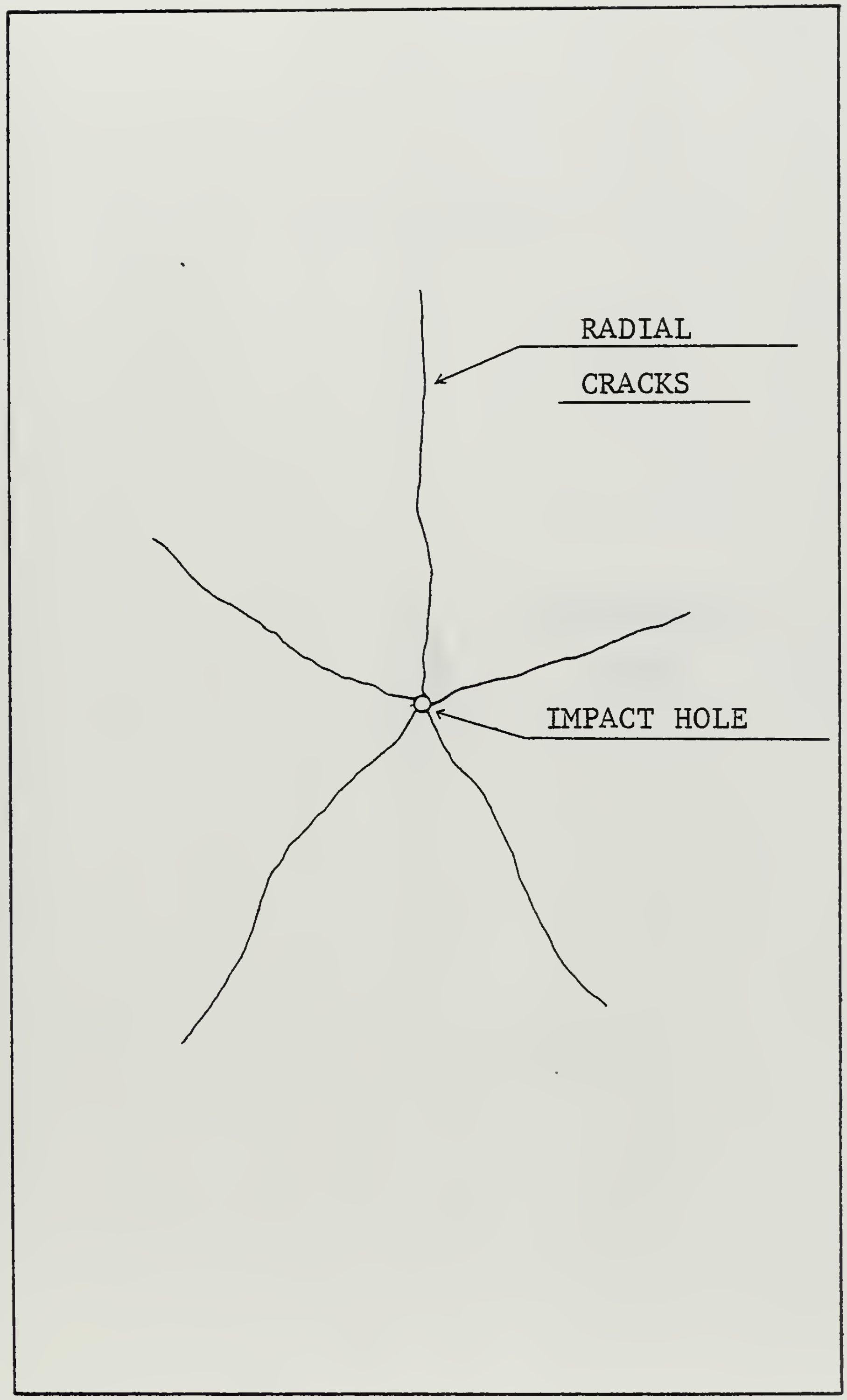

Figure II.1 RADIAL CRACK FAILURE

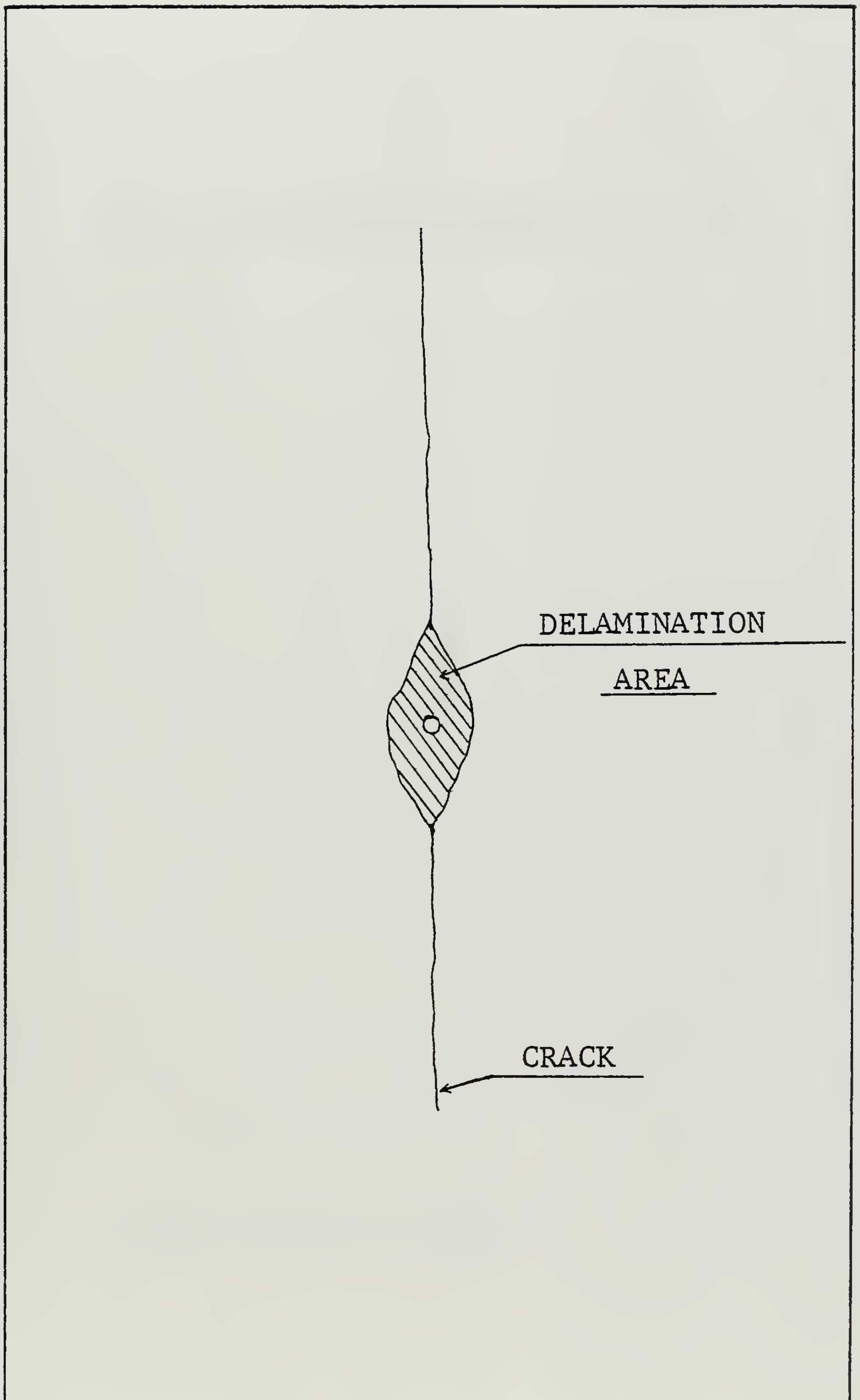

Figure II.2 LENGTHWISE CRACK FAILURE

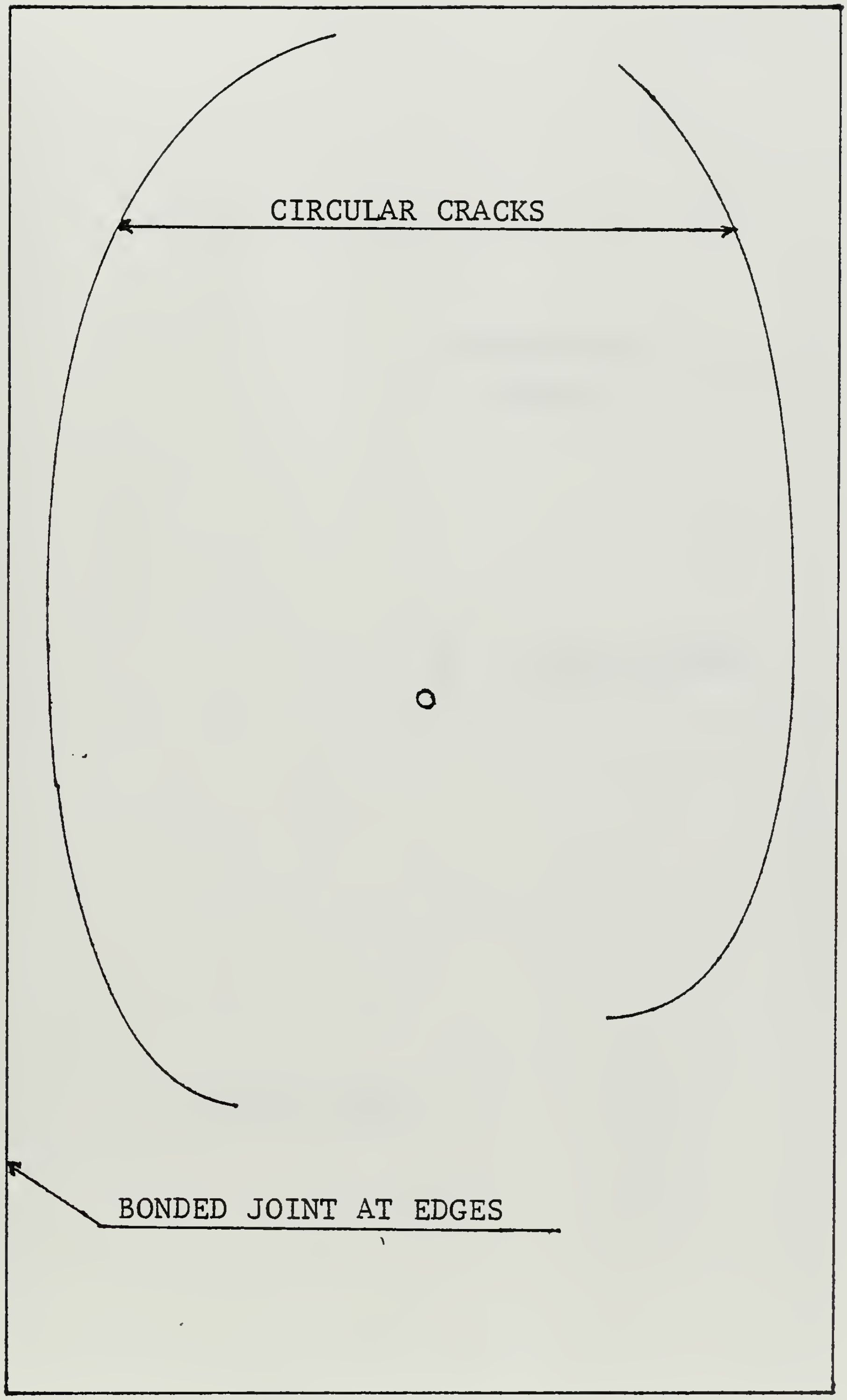

Figure II.3   CIRCULAR CRACK FAILURE

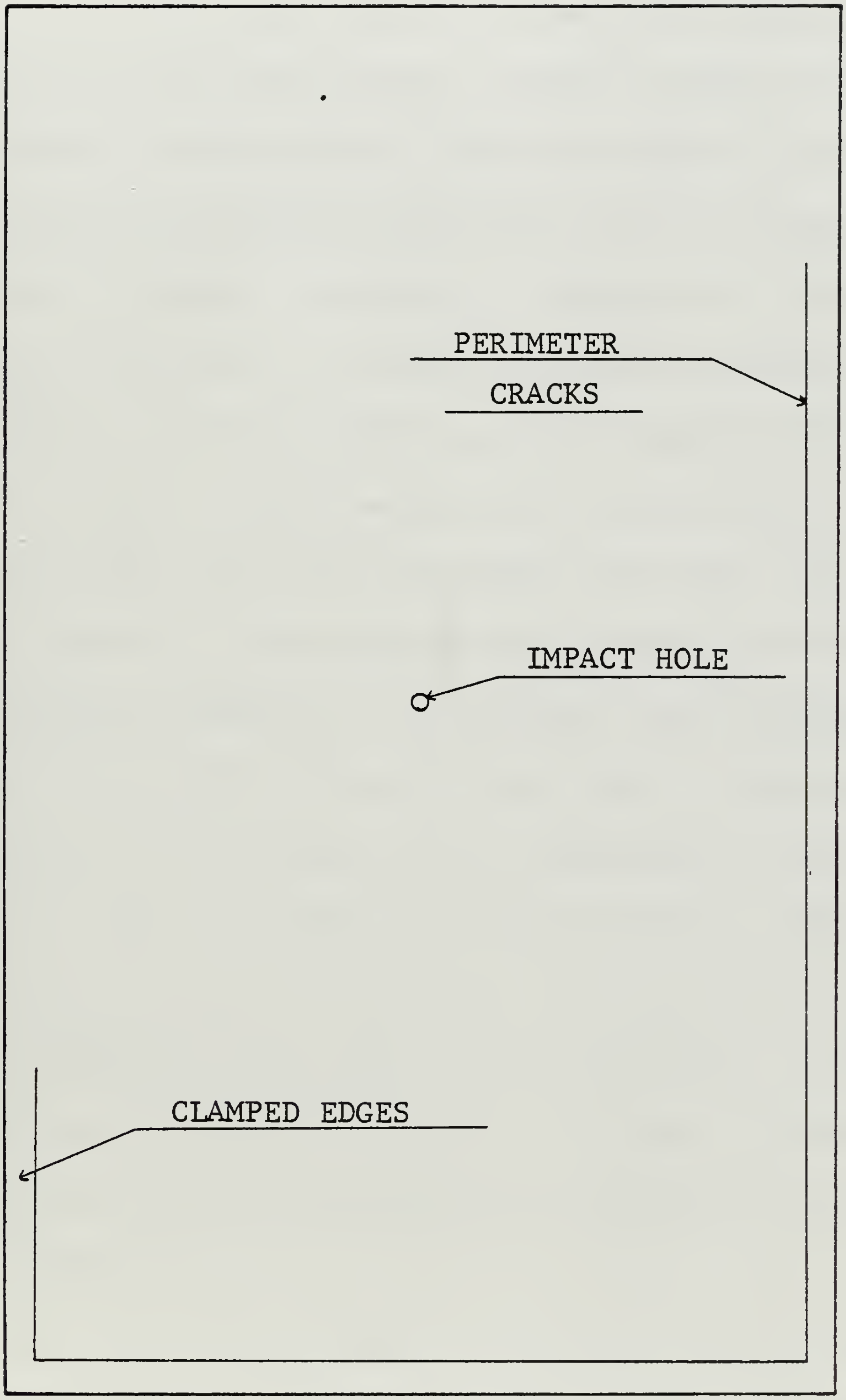

Figure II.4 EDGE LENGTH CRACK FAILURE

A fifth possible fracture type has also become apparent as a result of recent tests performed by Ultrasystems, Inc. [Ref. 9] for the Naval Weapons Center at China Lake, California. Ultrasystems studied the effect of hydraulic ram loading on buffering materials used in conjunction with stiffened aluminum test panels. Although the buffering material was used in an attempt to absorb sufficient energy to prevent catastropic failure of the aluminum walls, large radial cracks in the aluminum skins and pull-out of the aluminum skins from their fasteners was still the result. Whole lengths of the aluminum skins pulled out or unzipped from their attachments to stiff beams, which represented spars and ribs in an aircraft fuel tank, in the test. This unzipping of the material at the attachments is the fifth type of fracture and has occurred in aluminum skins. If this type of fracture also occurs in composites, the damage will probably be worse than it is in aluminum. A result of this type of fracture is the loss of wing box integrity, especially if much unzipping occurs along the spars, ribs and stiffeners.

The latter three fracture types are of particular interest here because they all lead to one conclusion; that since the major fractures were at the attachment sections

of the plate, the attachments are one of the critical areas of failure for a fuel tank of composite construction. An examination of the loading on the plate by the hydraulic ram also reveals that the attachment areas of the plate will be critical failure areas. The fluid pressures on the plate, created by the penetrating projectile, will produce three types of internal loads in the plate, membrane tensile and shearing forces, (N), thickness shearing forces, (Q), and bending and twisting moments, (M). This type of pressure loading, as seen in Figure II.5, can create high stress levels at the edges of the plate where the plate is restrained.

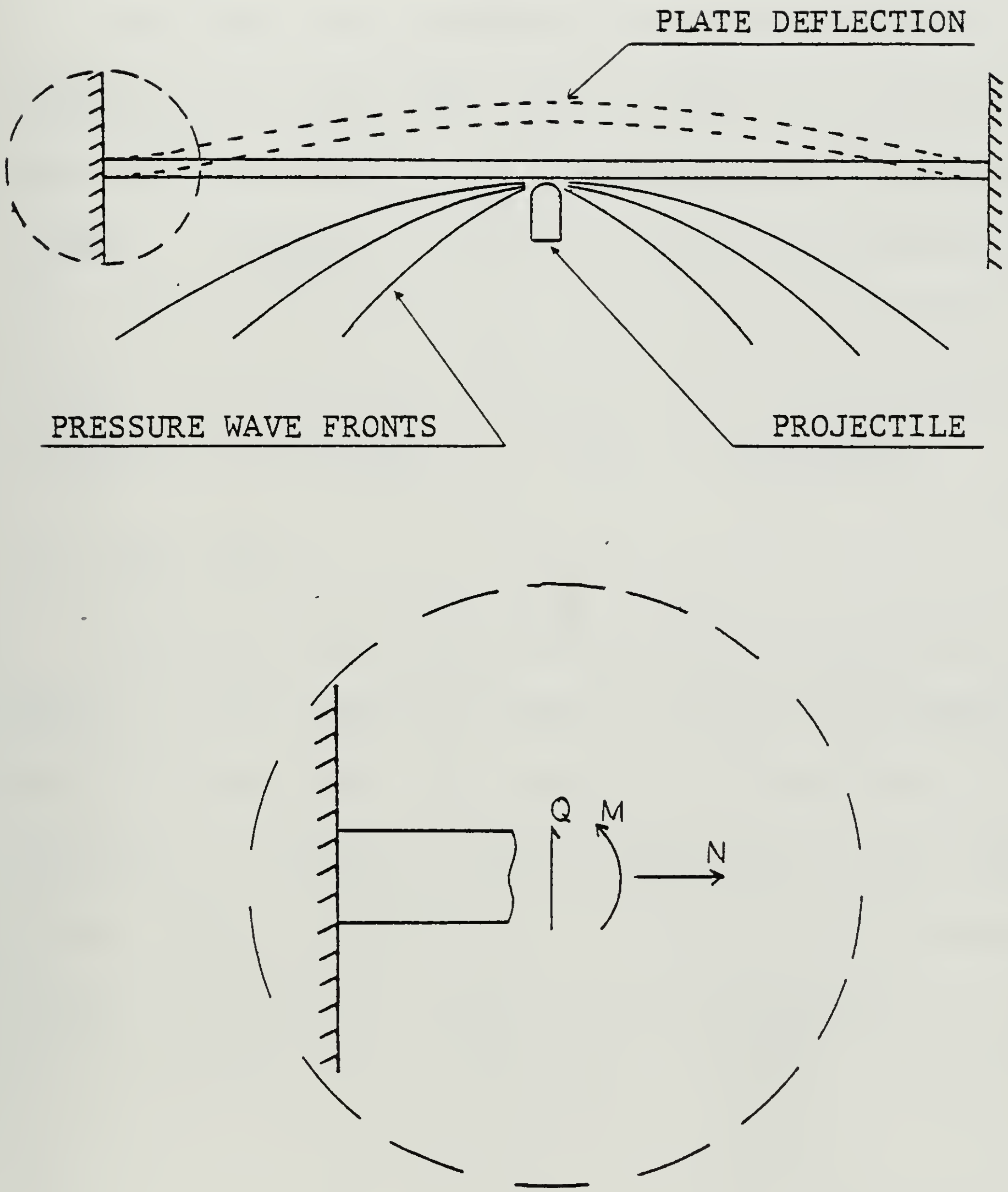

Figure II.5 HYDRAULIC RAM LOADING ON A PLATE

III. COMPOSITE FUEL TANK CONCEPTS

Many fuel tank parameters affect the hydraulic ram loading on the tank wall, such as depth of the fuel tank, tank wall construction, stiffener spacing, ullage, and others. Therefore, in order to better understand the effect of hydraulic ram loading on the total aircraft tankage, a study of the two main vulnerable areas of an aircraft, the wing and the fuselage fuel tanks, is necessary.

A. WING TANKS

Three different wing concepts were studied, the Conceptual Design of Navy Composite Wings (a delta wing design) [Ref. 10], the F-16 wing [Ref. 13], and the F-18 wing [Ref. 1]. The sections of interest in these wings are the fuel tank sections, where the thickness of the skin, the thickness of the wing, the distances between supporting members, and the lay-up of the skin are the important parameters. From these parameters, a fuel tank that is characteristic of current design concepts for military aircraft is developed as a representative wing fuel tank for hydraulic ram testing on a ballistic range.

The Navy's conceptual design for a composite wing, [Ref. 10], is concerned with the design of a wing for a V/STOL aircraft. In this concept, a 24-ply symmetric laminate wing skin of $\pm 60^{\circ}$ and 0° laminas of low modulus/ultra high strength graphite-epoxy was decided upon. In the fuel tank area, the spars vary from 24 to 52 inches apart, with rib separations varying from about 24 to 36 inches apart. The stringers, approximately 4 inches apart, add to the longitudinal stiffness of the structure. The wing tank is approximately 7 inches thick. This wing tank section is illustrated in Figure III.A.1.

The F-16 wing concept [Ref. 13], is more of a beam-like, straight wing design. In this design, a graphite-epoxy laminate of varying thickness, and therefore a varying number of plies, was decided upon. Over the fuel tank portion of the wing, the thickness of the laminate varies from 0.5 inches to 0.3 inches, with a symmetrical laminate layup of 0's, 90's, and ± 45's. Another F-16 wing skin concept is that of a graphite-epoxy-aluminum honeycomb, approximately 3/16th of an inch thick. The wing thickness varies from about 5 3/4 inches to 2 inches. Softening strips in the skin are also being considered to restrict any cracking that might occur. Whether or not softening strips will

restrict the cracking that occurs as a result of hydraulic ram loading is not known at this time. They have been found to be effective in restricting the propagation of cracks due to fatigue. The ribs and spars are of aluminum, and no stiffeners are used. Distances separating ribs, spars and stiffeners are still unknown. This design is much different from that of the Navy V/STOL design concept, but is quite similar to that of the F-18.

The F-18 wing [Ref. 1], shown in Figure III.A.2, is also a beam-like, straight wing design. A graphite-epoxy laminate of low modulus/high strength is being considered for the skins. Only one large composite panel is used as the skin on each surface of the wing over the fuel tank section. The thickness of the wing tank varies from 9 inches to 5.5 inches, and it has aluminum spars and ribs, with no stiffeners. The spars are approximately 9 inches apart from each other, and the ribs are approximately 25 inches and 57 inches apart. At the present time, the McDonnell-Douglas Aircraft Corporation [Ref. 1] is not planning on using either softening strips or a honeycomb type of skin for the F-18.

These three wing concepts have been compared to each other in order to develop a reasonable and realistic concept

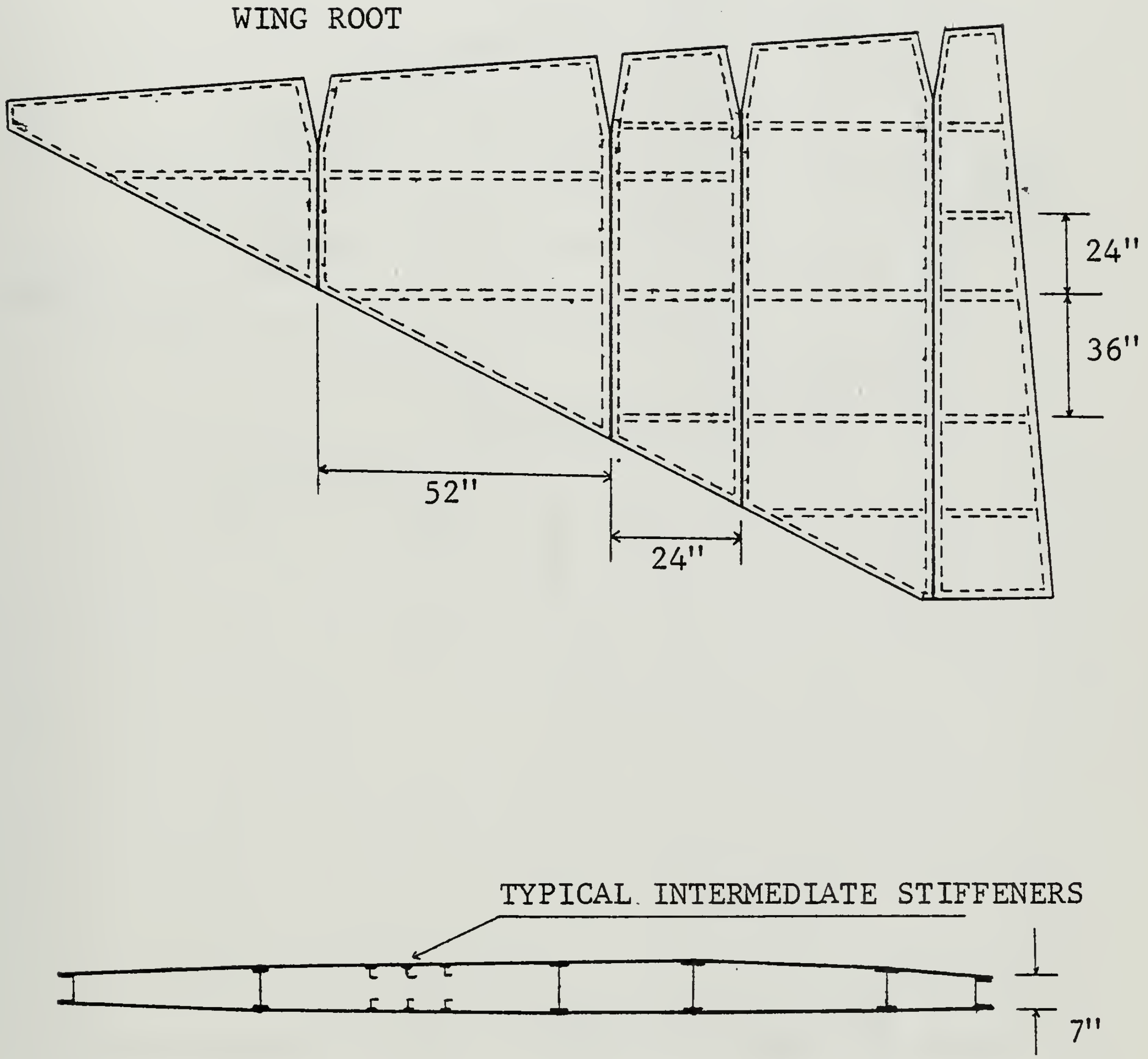

Figure III.A.1 V/STOL WING CONCEPT

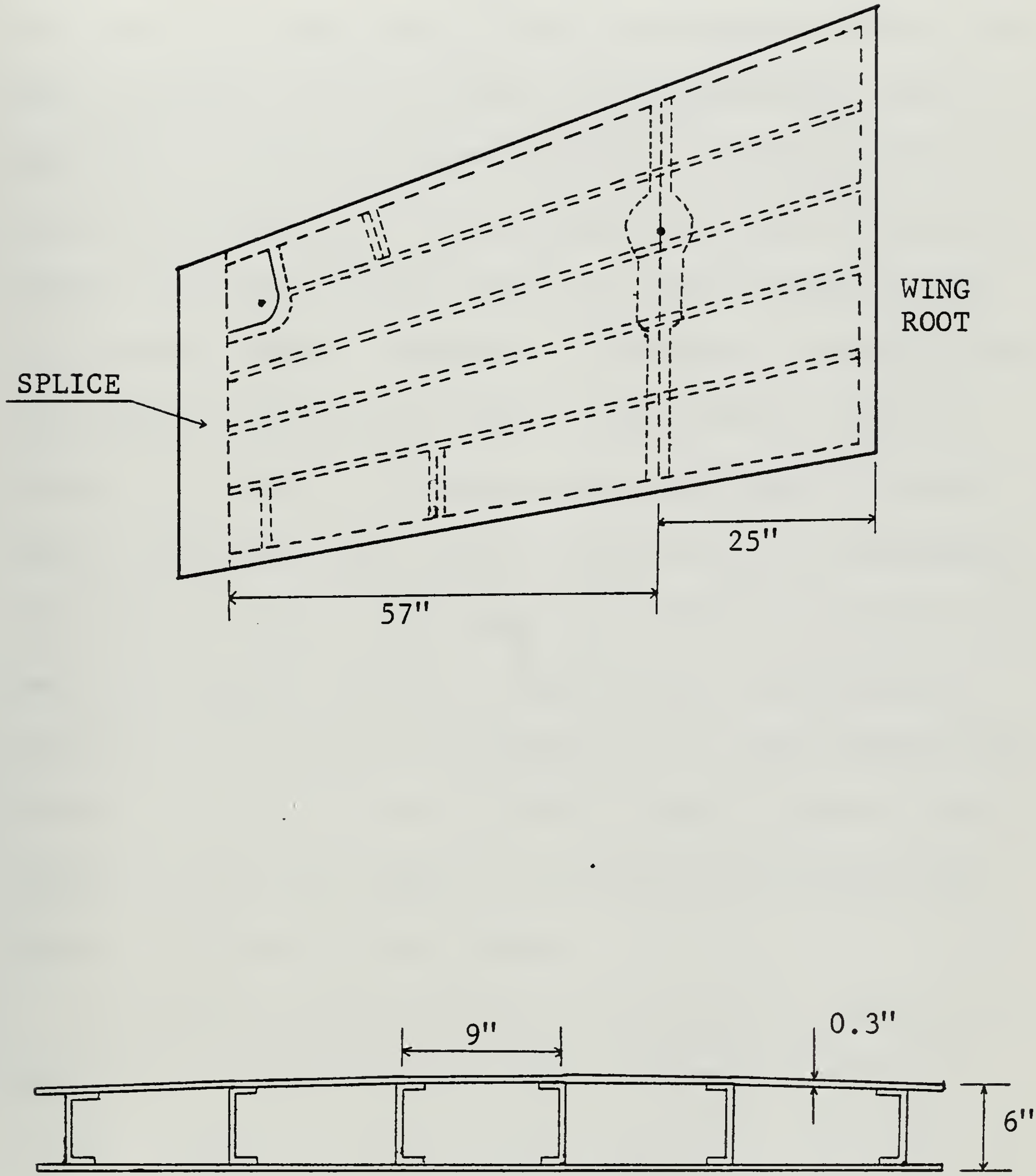

Figure III.A.2 F-18 WING CONCEPT

of a typical wing fuel tank. This permits hydraulic ram testing with a tank that not only approximates actual wing tank concepts, but also allows for consistencies that are necessary in any experimental analysis. A typical section proposed for hydraulic ram testing is one that is approximately 24 inches between ribs and 8 inches between spars, or 4 inches between stiffeners. Therefore, by using plates 24 inches by 24 inches in size, a test section that is similar in size to the proposed aircraft wing fuel tanks can be developed. The section should also be approximately six inches thick to approximate the thickness of actual wings and thus present a similar liquid volume. This typical section is illustrated in Figure III.A.3. One disadvantage of this size of test tank is that it may not give an indication of the total amount of unzipping that might occur because of hydraulic ram loading.

A problem that has severely restricted the testing of this type of section at the Naval Postgraduate School is cost. A graphite-epoxy composite laminate that is 24 inches square and ½ inch thick would cost between \$1500 and \$2500 to produce. It would require outside contracting for its construction since there are only limited composite manufacturing capabilities at the Naval Postgraduate School.

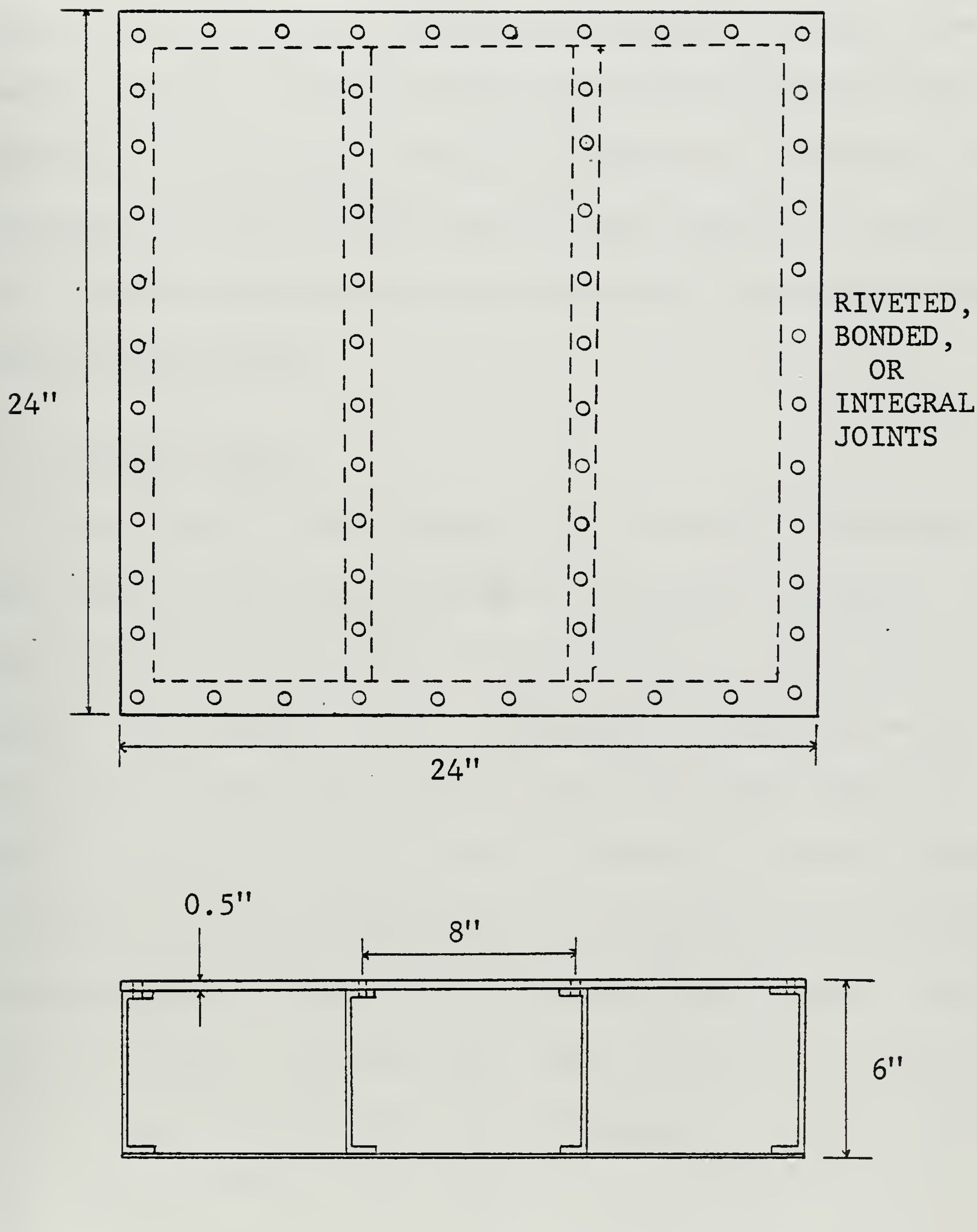

Figure III.A.3 WING TANK TEST SECTION

Although that is a relatively high price for hydraulic ram testing, it is inexpensive compared with the testing and destruction of at least one complete wing. It also allows better control of the variables that would greatly affect hydraulic ram testing, such as the material selection, the attachment type, and the layup of the composite. The use of a common test cell also permits better data correlation between experiments.

B. FUSELAGE TANKS

The hydraulic ram loading in an aircraft's fuselage fuel tanks creates some different problems from those associated with the hydraulic ram loading in wing fuel tanks. For example, since a fuselage fuel tank normally has greater fluid depth than a wing fuel tank, the projectile has more time to tumble, and hence a greater amount of energy of the projectile may be dissipated in the fluid, increasing the effect of the hydraulic ram loading. There will be greater pressures and loads in the tank, and one could predict potentially greater damage to the fuel tank as a result. Since more energy is dissipated into the fluid, it is also possible that the projectile might not have enough energy to penetrate the rear wall of the fuel

tank. This would eliminate any additional projectile damage to the interior of the aircraft, but would not eliminate the possibility of catastropic failure of the rear wall of the fuel tank caused by the hydraulic ram loading.

Not only do the fuselage tanks contain a greater depth and volume of fuel, they may also be semi-protected by various aircraft systems such as engines, landing gear, or electronic equipment. These other systems could either reduce the damage to the fuel tank, or increase it. A fuel tank in the outboard section of the fuselage might also be next to the engine or the air inlet to the engine, which could create fuel ingestion problems if hit by a projectile. A catastropic failure of the fuel tank in one of the above areas, even if caused by a small caliber projectile, may result in the complete loss of the aircraft. There may also be fluid on both sides of the fuselage tank whereas there may not be on the wing tanks. This also presents a problem because it could mean damage not only to one fuel tank but to many, and possible rapid loss of a large percentage of the aircraft's fuel supply. Lastly, fuselage fuel tanks generally have highly curved surface skins for the wall that is most likely to be impacted by a projectile.

A number of fuselage fuel tank concepts are currently under consideration for various aircraft, [Ref. 11] and [Ref. 13]. In general, particular sizes of the fuel tanks, such as distances between longitudinal stiffeners and distances between ring frames, are unknown. There are two main composite skin concepts under consideration for current and upcoming aircraft. The first concept is one of a graphite-epoxy laminate skin, with varying thickness from about 0.125 inches to about 0.3 inches, [Ref. 13]. These sheets are very similar to those designed for the wing skins, having a layup of 0's, 90's, and $\pm$45's, although specifically designed for the fuselage loading conditions. These skins are usually highly curved, whereas the wing section skins are only slightly curved. The plate sizes vary from approximately 12 inches to 45 inches wide, and can extend around the fuselage from the top to the bottom, or they may have less lengthy sections.

The second concept is that of a honeycomb skin, with a thin graphite-epoxy laminate outer skin, from 0.04 inches to 0.06 inches thick, with aluminum for the honeycomb, [Ref. 13]. The total skin thickness would be no more than 0.25 inches thick. Bladders and rigidized reticulated foam are also being considered for fuel containment, energy absorption, and resealing capability.

For hydraulic ram testing of a fuselage fuel tank, a 24-inch by 24-inch by 18-inch cube could be expected to closely approximate an actual fuselage fuel tank. It allows for a realistic representation of volume and depth characteristics in an actual tank. In much the same way as with a wing tank testing section, the fuselage tank testing section should permit experimentation with different methods of attachment, and different types of skins. An example of the proposed test tank is illustrated in Figure III.B.1. Although the concept is drawn with a flat skin, a test section using curved skins would permit the investigation of the effects of curvature on the failure of composite walled fuselage fuel tanks. A flat walled tank is sufficient to explain the effects of fluid depth and volume, and also to give a basic idea of the type of fracture to be expected in a fuselage fuel tank. If bonded or integral joining techniques are to be investigated, more than one test tank would need to be built. Construction costs would probably be only slightly lower than the costs of constructing a wing test tank, mainly because of reduced composite material needs, but they would still be relatively inexpensive in comparison with the testing of an actual aircraft's fuselage fuel section.

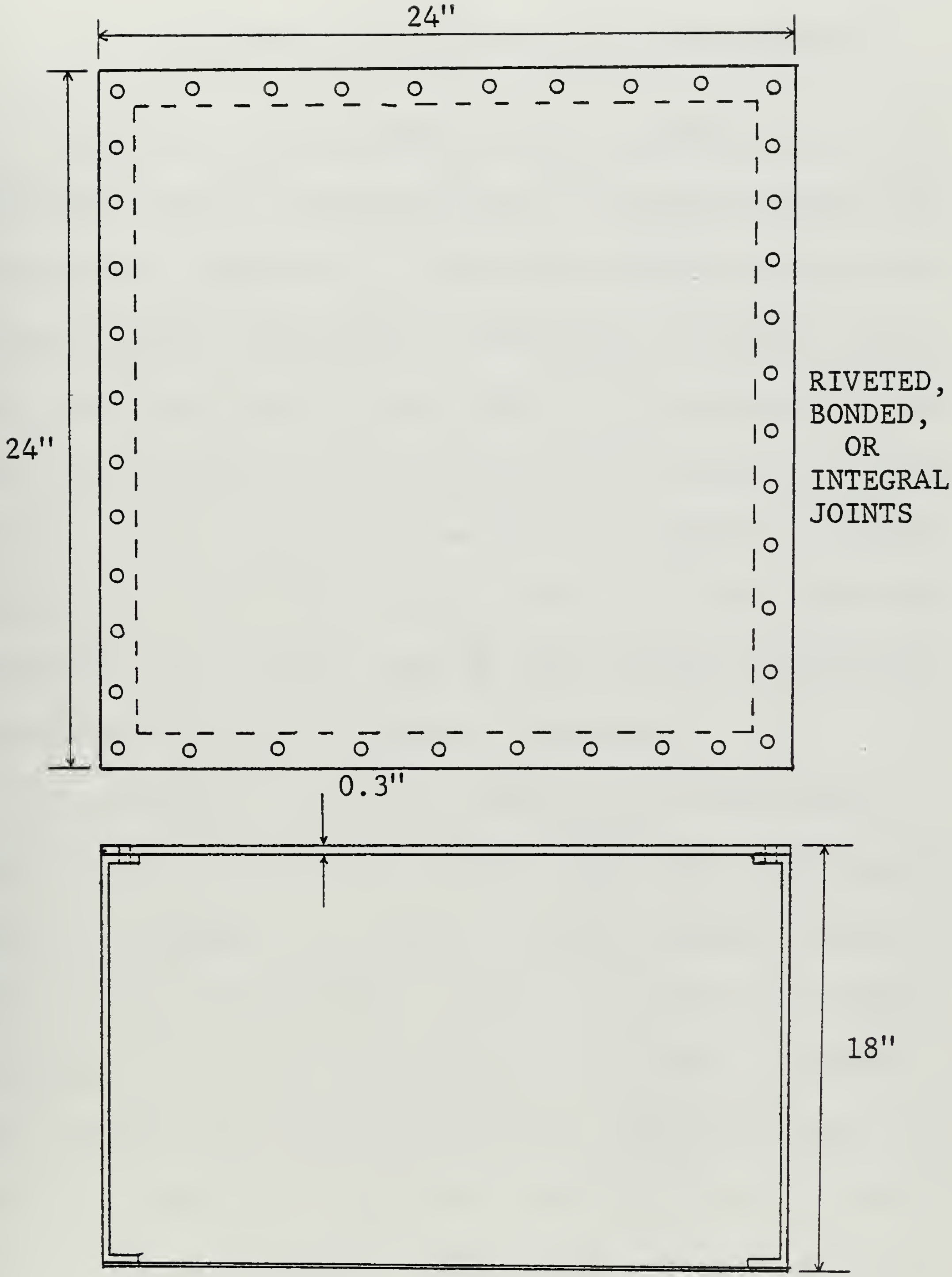

Figure III.B.1 FUSELAGE TANK TEST SECTION

IV. SELECTION OF A TYPICAL JOINT FOR ANALYSIS

Two main joining techniques are currently in use for laminated fibrous composites such as graphite-epoxy and boron-epoxy composites. The first technique is that of bonded joints; the second, mechanically fastened joints. Each technique has its advantages and disadvantages, and each has a number of variations as to its method of application. Since it is believed that the joints or attachments may be the most critical part of a fuel tank under hydraulic ram loading, their design and use in aircraft structures becomes of primary importance.

In aircraft structures, bonded joints are mainly restricted to single lap joints, [Ref. 2]. The composite skin, for example, is bonded to another member, such as a spar or rib of aluminum, in order to provide high shear strength. A bonded joint behaves in a manner similar to an aluminum joint, and is therefore reasonably easy to analyze, even for a transverse shearing force, as would be created by the hydraulic ram loading. It also provides slightly higher lap shear strengths than a mechanically fastened joint. Four failure modes exist for bonded joints,

[Ref. 2]. These four failure modes are: axial failure of the laminate, interlaminar shear failure, adhesive failure of the adherent/adhesive interface, and cohesive failure of the adhesive. A failure involving one of the first two modes indicates that the joint was reasonably sound, with failure in only the laminate, and is a very desirable quality for bonded joints. A failure involving one of the latter two modes indicates that the joint failed. This is not desirable, since it usually means a loss of structural integrity and possible fuel flow to vital areas of the aircraft.

Two characteristics of bonded joints seriously limit their use in aircraft. They prevent routine inspection of underlying spaces, and they do not allow easy replacement of the laminate. A section with bonded joints usually means that there is no need to inspect the underlying area, or else it can be inspected from another area. There is some thought of using bonded joints, but only for fuselage skins and fuel tanks, where the need for inspection of the structure is not as great as that needed for wing structures. Thus mechanically fastened joint designs are usually preferred (at least at present) for composite laminate usage in aircraft.

Mechanically fastened joints not only allow for inspection, but they also are easier to manufacture and they permit relatively easy replacement of the laminate. Four failure modes exist for mechanically fastened joints, [Ref. 2]. The first three, shearout of the material, tension failure of the material, and bearing failure of the material, all indicate a material failure of the joint in which the fastener was properly designed. The last mode of failure is one of bolt or pin failure, which is highly undesirable for a joint since it indicates an unnecessary loss of structural integrity. These are the expected modes of failure of the joint under normal aircraft flight loads, but they do not apply when hydraulic ram loading occurs. In this case, the critical load is a combination of the bending moment, the transverse shearing force, and the tensile force loads on the joint.

The current method of attachment, using mechanically fastened joints, is usually with some type of bolt [Ref. 1] and [Ref. 13]. Some bolts are removable, while others are not, depending upon the need for accessibility of the underlying area and the necessity for field replacement of the material. One current design concept is to use high-lock pin collar bolts of quarter-inch diameter, with a bolt

spacing of five diameters, [Ref. 1]. The use of this type of fastener not only requires the drilling of a hole in the composite plate, which will create stress concentrations around the hole, but it also requires a channel groove or an "O"-ring to be built into or cut into the material in order to keep the fastener head flush with the surface of the composite skin. A filler would then be used to fill the groove, resulting in a smooth surface. This groove only increases the problem of understanding what happens at the joint. Unknown stress concentrations and reduced capability of the joint are the results. Figure IV.1 shows how the composite laminate appears at the joint. The most widely accepted method of attachment is that of a combination of removable and non-removable fasteners. One proposed design is a line of removable fasteners, interspaced with non-removable fasteners, which reduces the accessibility of an area under a composite panel.

Since the use of mechanically fastened joints is the most widespread among the types of fasteners used by the aircraft manufacturers, this study will restrict itself to their usage. The actual load concentration factors become of primary interest in the study of mechanically fastened joints for composites, since a hole through which to insert the fastener is needed to use it.

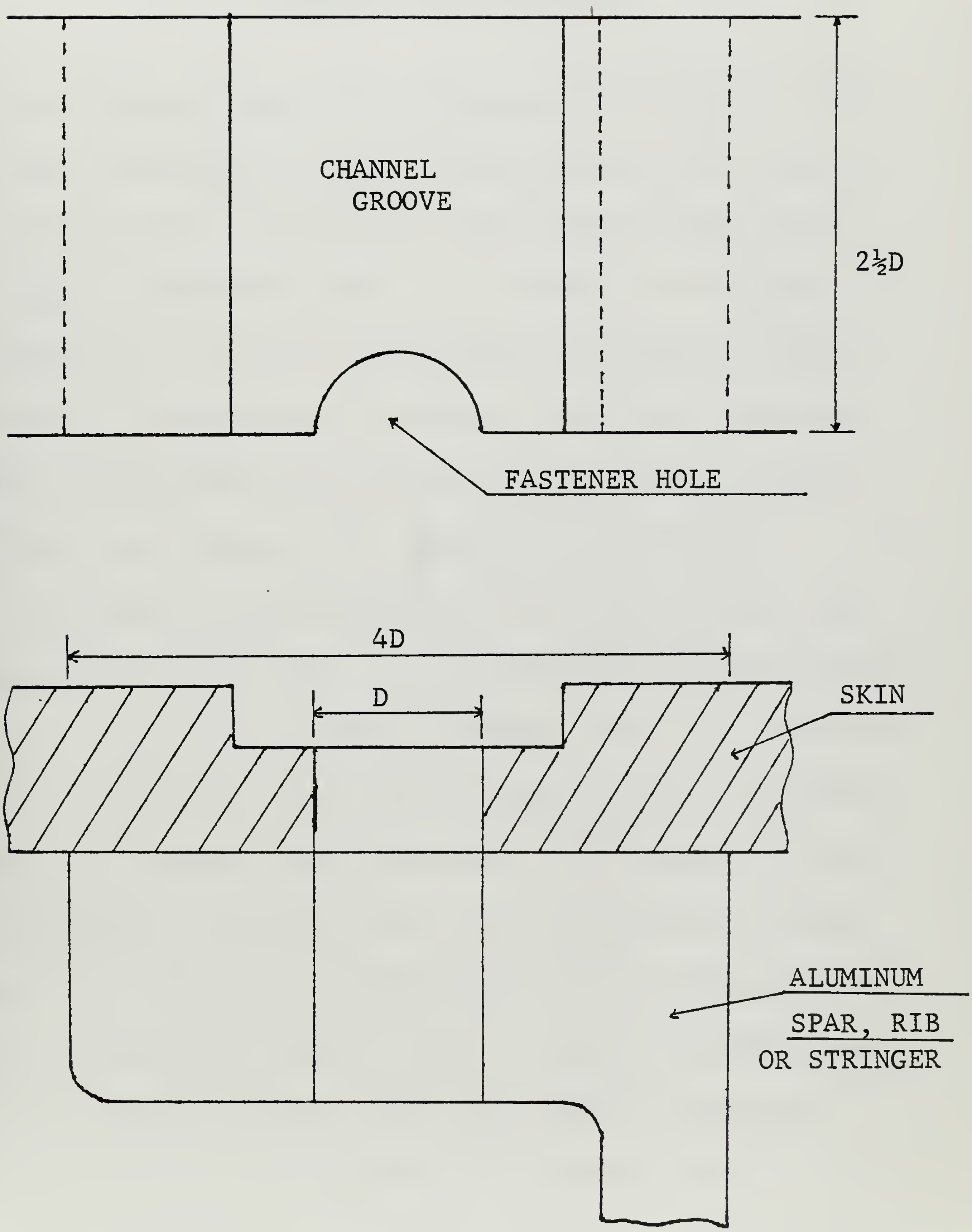

Figure IV.1 TYPICAL MECHANICALLY FASTENED JOINT
(HALF SECTION)

V. LOAD MAGNITUDES IN THE FUEL TANK WALL DUE TO HYDRAULIC RAM LOADING

The internal loads in an aluminum fuel tank wall that has been penetrated by a 12.7 mm projectile and subjected to hydraulic ram loading have been obtained with the use of computer programs, [Ref. 4]. Although these loads are approximations based upon the predicted pressure loading produced by hydraulic ram loading on the fuel tank wall, general magnitudes of the tensile force, bending moment, and transverse shear at the edge can be obtained.

The loads are time dependent, but two different load trends can be identified. Q, the thickness shearing force, is the first load to assume a dominant role. It increases very rapidly, while the tensile force, N, and the bending moment, M, increase very gradually. In reaching its maximum of nearly 700 pounds/inch, Q is substantially larger than N or M, where the tensile force is approximately two orders of magnitude smaller than Q, and the bending moment (in inch-pounds/inch) is about one order of magnitude smaller than Q. The skin was 0.125 inches thick.

The slow increase in the magnitude of N with time is the second load trend. Taking about ten times the amount

of time that it took Q to reach its maximum, N increases until it reaches a magnitude of about 1500 pounds/inch. As N increases to this load level, Q decreases until it is about an order of magnitude smaller than N. The bending moment, M, is about the same magnitude as Q.

Although N and M remain at about the same order of magnitude with increasing distance from the point of impact, Q does not. Q increases with increasing distance from the point of impact of the projectile. Therefore, Q can be either much larger or much smaller than the indicated magnitude, depending on the distance of the point of impact from a supporting member in the fuel tank.

Although it is not the largest load that the wall must support, Q appears to be a major factor in the internal loading of the fuel tank wall due to hydraulic ram. Not only is its magnitude significant, but its rapid rise to that level may also be a contributing factor in understanding the joint failures produced by hydraulic ram loading. M also becomes very large, although always less than Q. N does become the largest of the three loads, but only after a considerable amount of time.

VI. ANALYSIS FOR STRESS CONCENTRATIONS AT THE JOINT DUE TO UNIT TENSILE, SHEAR, AND BENDING MOMENT LOADING

A computer study of a mechanically fastened joint of aluminum has been undertaken to investigate the stress fields that are in the aluminum plate around a fastener as a result of the restraint of the fastener in the joint. A finite element program was used to analyze the joint (Figure VI.A.1). The computer analysis was accomplished with the plate elements and the membrane elements provided by SAP IV, a multi-element, finite element structural analysis program [Ref. 6]. The purpose of this analysis was to investigate the stress concentrations that would result from a tensile force/unit length, N, a bending moment/unit length, M, and a transverse shearing force/unit length, Q, acting along the edge of the joint as shown in Figure VI.A.2.

A. PLATE MODEL

Since the area of the joint is small, and since it is symmetrical, only 56 plate elements and 73 nodes were used to model one-half of the joint. Figure VI.A.1 shows the joint model with the layout of the nodes and the numbering

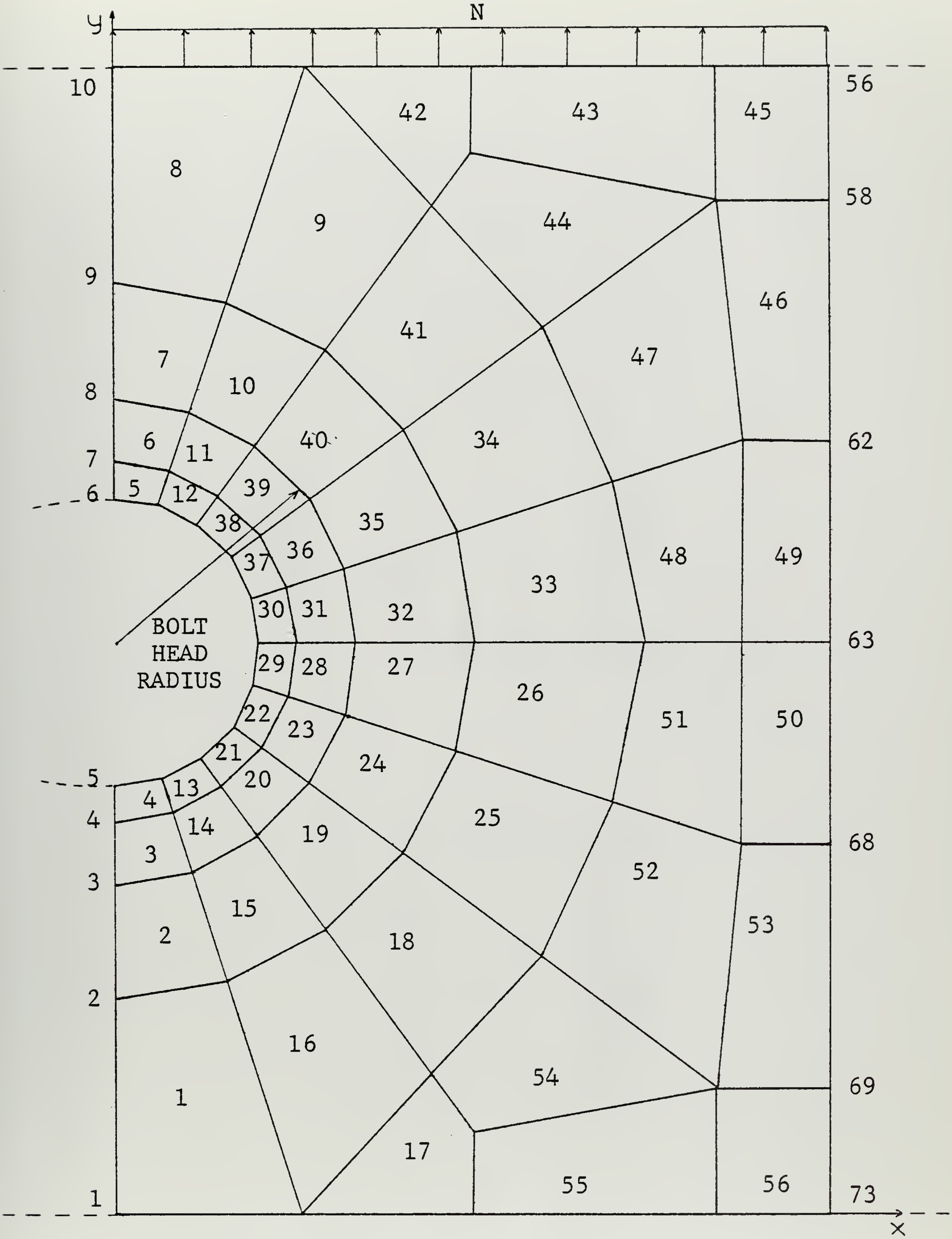

Figure VI.A.1 PLATE MODEL

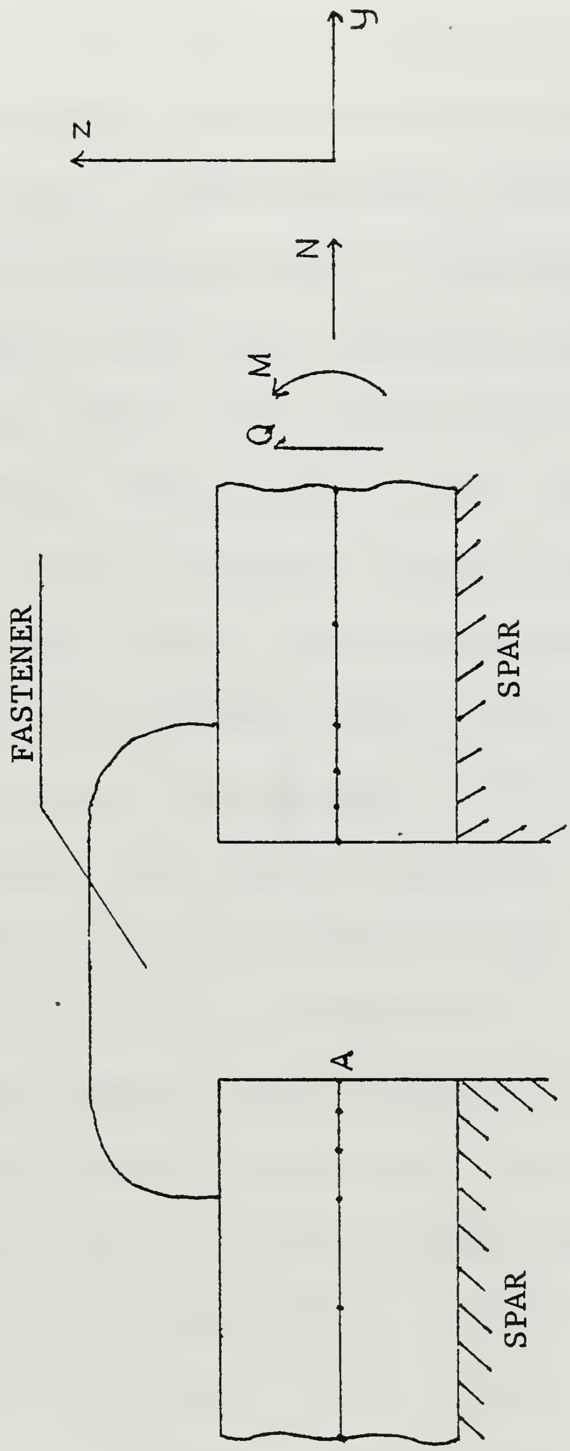

Figure VI.A.2 JOINT LOADING

of the elements. Unit loads/unit length were used so that actual stress resultant concentration factors around the hole would be found. A two-dimensional thickness model is analyzed in the following section to determine the stress concentration due to the stress resultant concentration.

The intricate part of this analysis was in defining the boundary conditions. The problem is not simply a plate with a hole, but is a joint in which a fastener constrains the displacement of part of the plate material and therefore picks up some of the internal loads in the plate.

It was assumed that the fastener would not fail, but would remain rigid. The skin was permitted to move in the y-direction except where the fastener would restrict such movement, such as at point A, Figure VI.A.2. It was also assumed that the fastener would restrain the movement of the plate in the z-direction where the fastener head overlapped the plate, and it was also assumed that the fastener did not permit any rotations where it overlapped the plate. The boundaries of the model, nodes 1 through 10, and nodes 56, 58, 62, 63, 68, 69 and 73 were not permitted displacements in the x-direction or rotations about the y-axis, due to the symmetry of the problem. The edge between nodes 10 and 56 was where the unit loads/unit length were applied,

and the edge between nodes 1 and 73 was left free to displace in the y-direction.

Figures VI.A.3, VI.A.4, and VI.A.5 show how the stresses in the plate, Sxx and Syy, varied around the hole for a unit tensile force/unit length. Although the joint is somewhat similar to a plate with a hole, the fastener was available to resist the tensile forces put on the plate; hence, higher stress concentrations than those of a plate with a hole in it were the result. Since only the fastener restricts deflection in the y-direction, all of the tensile force is picked up at the fastener, causing stress concentrations near five in compression in the plate.

Not only were there large stress concentrations in compression where the fastener picked up the load, but there were also large stress concentrations in tension in the y-direction of approximately 5.5. Since there is a hole, and the loads will not be picked up until they get around the hole, large stress concentrations will exist even where the fastener has no effect in restricting deflections in the y-direction. Results of the analytical studies of Howland and Theocaris [Ref. 2] indicate that stress concentrations near five or five and a half are quite close to

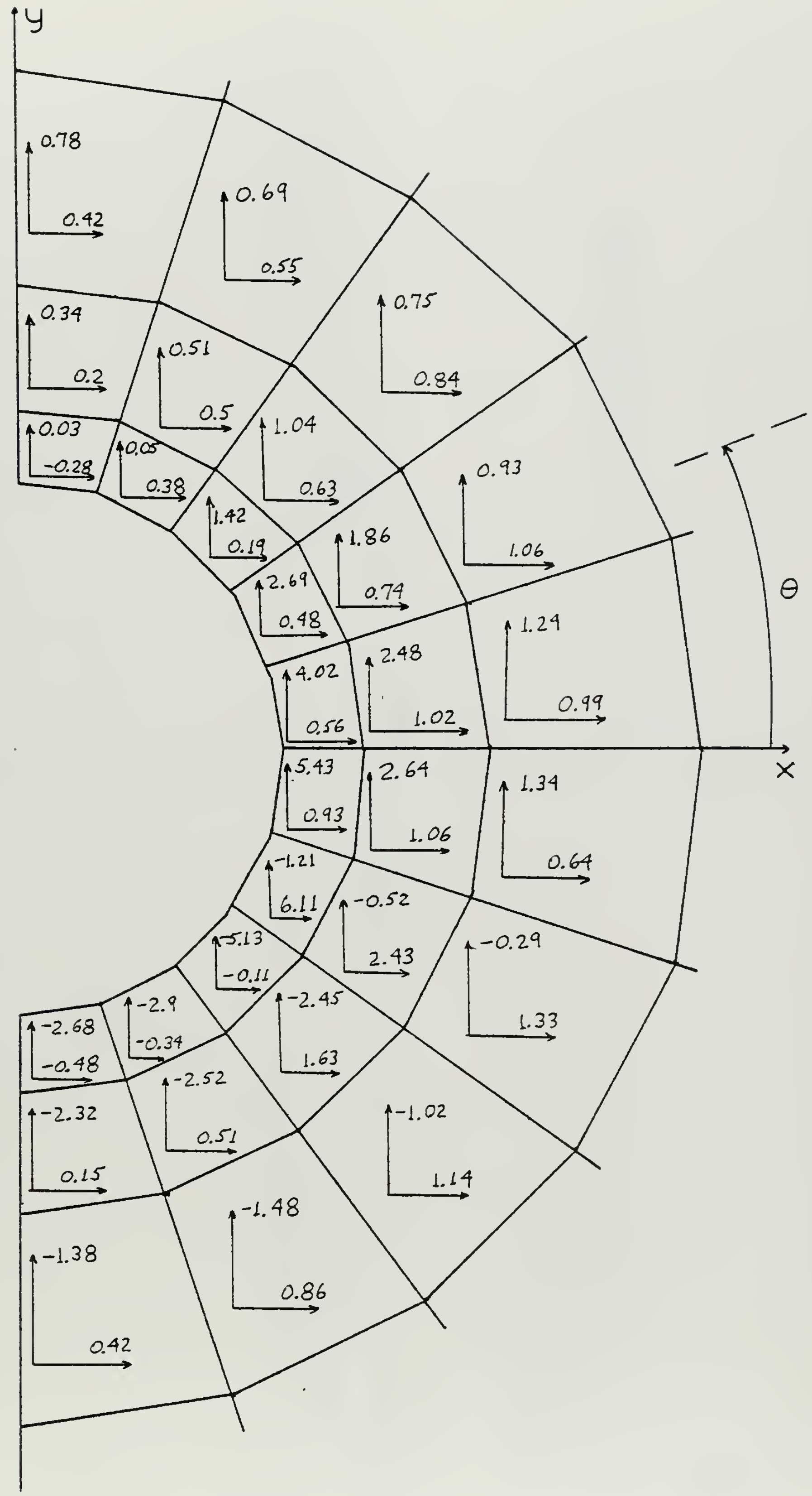

Figure VI.A.3 STRESS FIELD AROUND HOLE DUE TO N

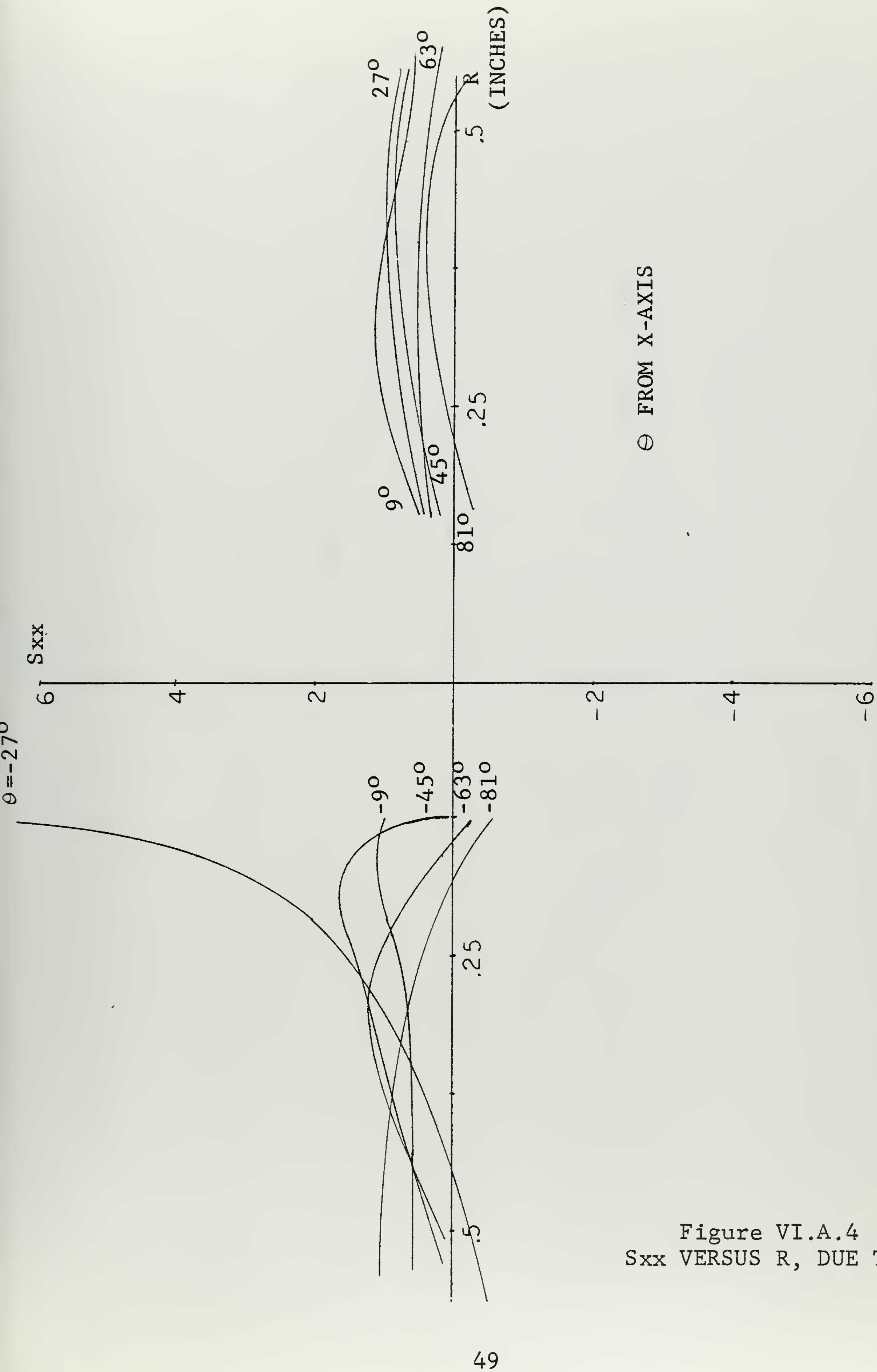

Figure VI.A.4
Sxx VERSUS R, DUE TO N

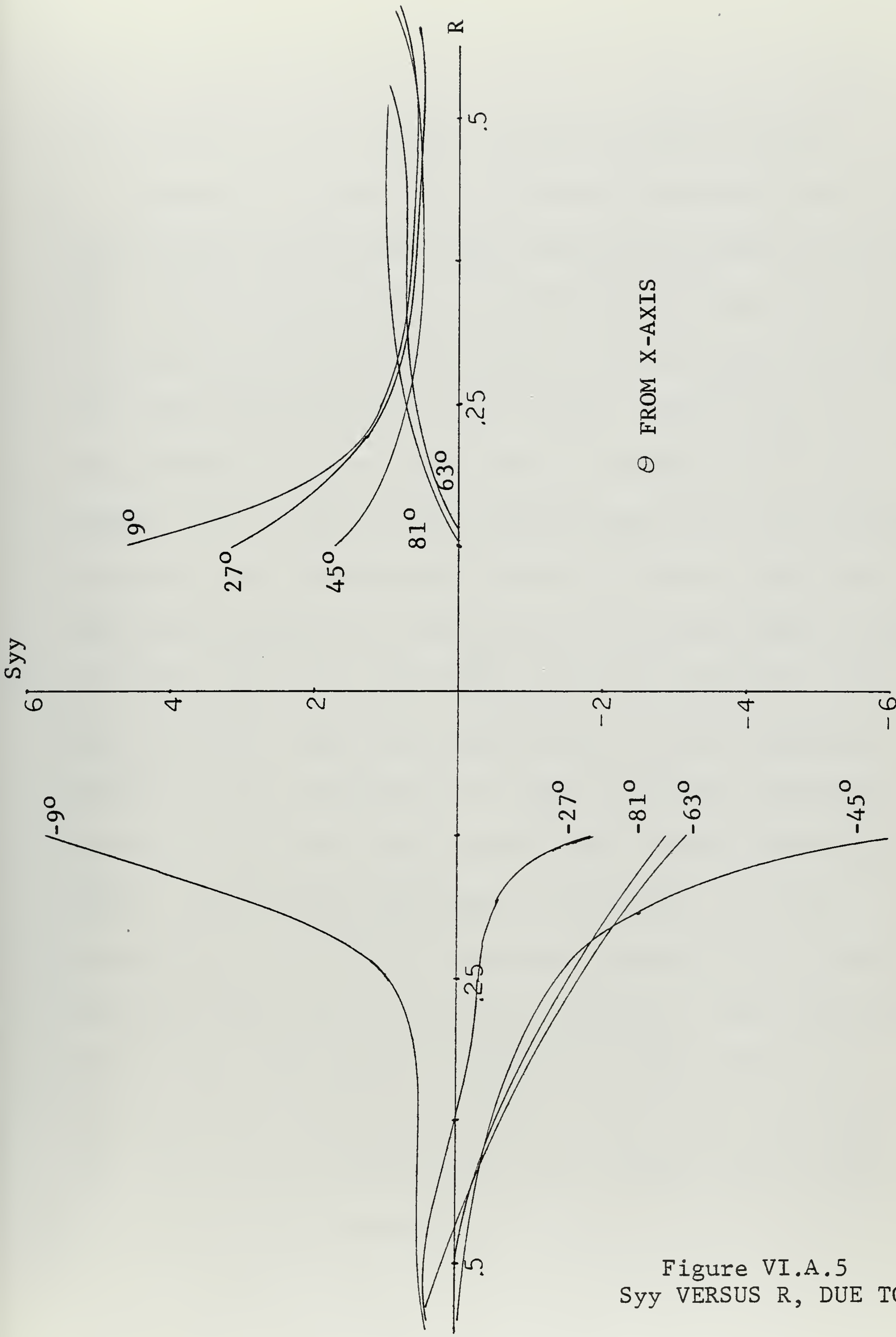

Figure VI.A.5
Syy VERSUS R, DUE TO N

their results on mechanically fastened composite joints, as presented in the Advanced Composites Design Guide.

Figures VI.A.6 and VI.A.7 show how the moments due to the applied bending moment and the applied transverse shearing forces, respectively, are distributed through the plate. No moments were found in the section of the plate where the bolt head was influencing the material rotation, as one would expect. The important idea to remember about the moment field is that if the bending moment and the thickness shearing forces are the most critical loads in the joint, then these figures will give only a very basic idea of what one would expect for the bending stresses around the hole. Unfortunately, since the program does not give thickness shearing stress resultants for either the unit transverse shearing force/unit length or the unit bending moment/unit length (only moment resultants are printed) it is not possible to determine the thickness shearing force concentrations around the bolt hole for these two loads using the plate model. The way one can determine the transverse shearing stress state around the hole for these two loads is to use a three-dimensional finite element model of the joint using brick elements.

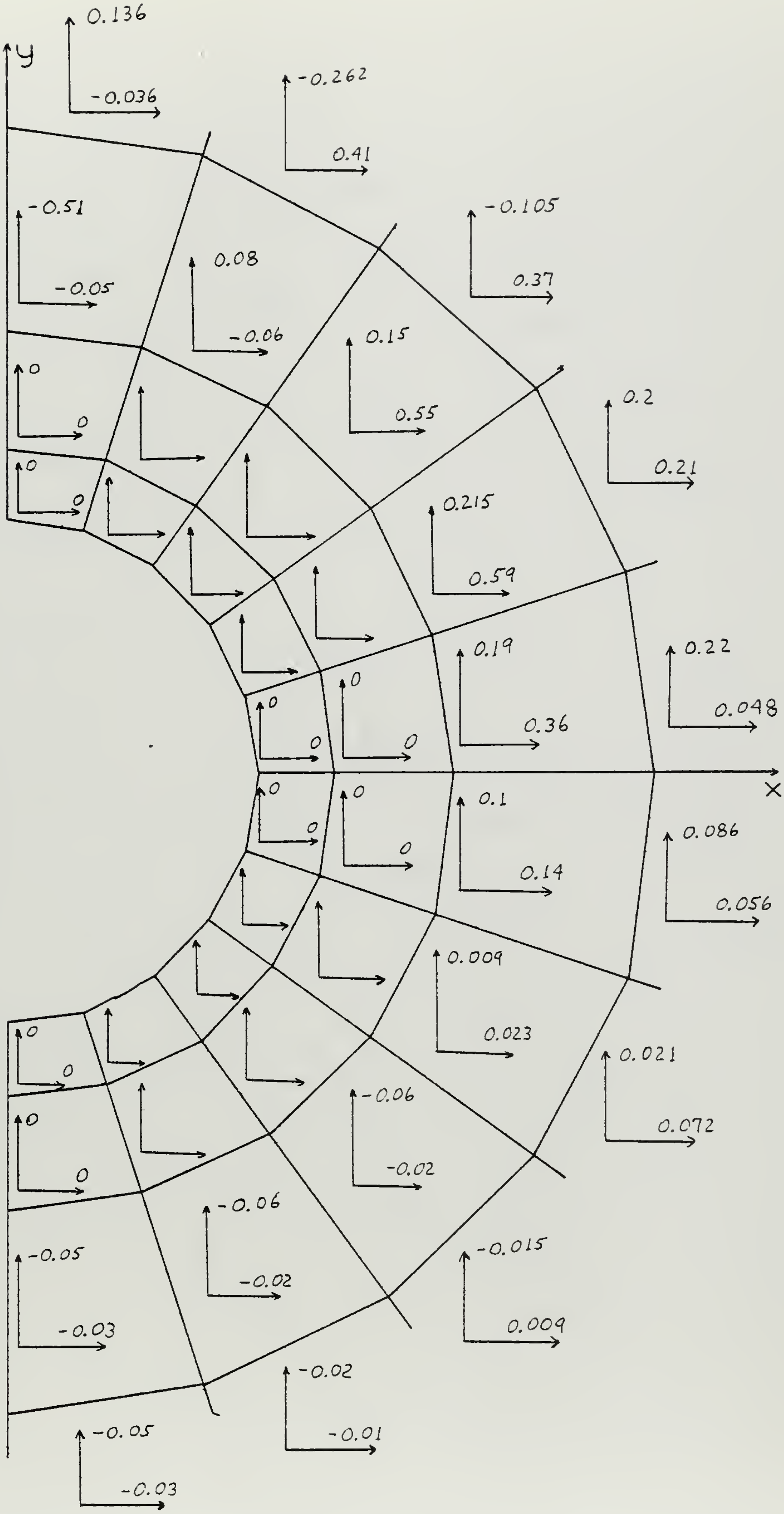

Figure VI.A.6 BENDING MOMENT FIELD DUE TO BENDING MOMENT

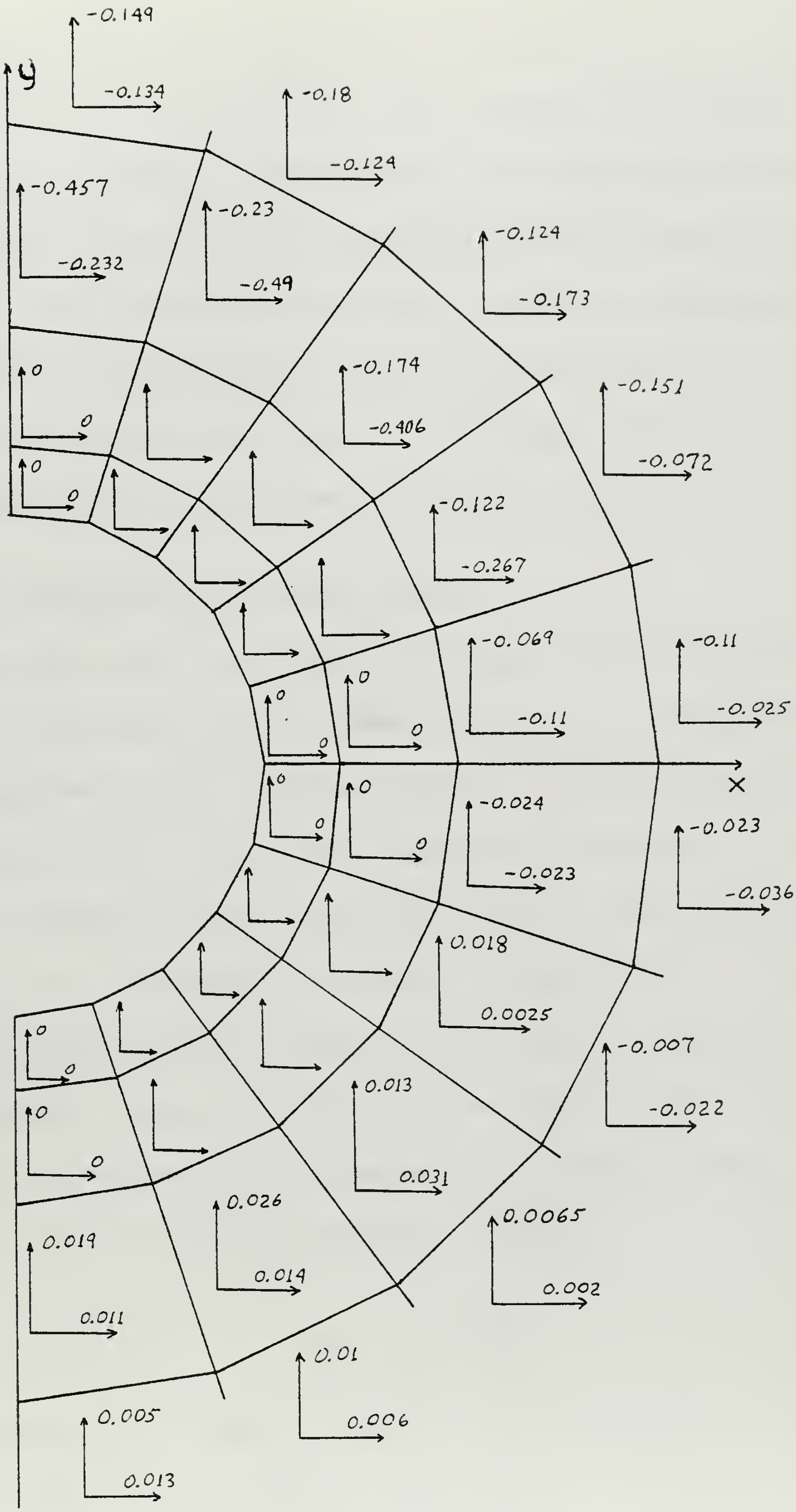

Figure VI.A.7 BENDING MOMENT FIELD DUE TO TRANSVERSE SHEAR

Since the stress resultant concentrations due to these two loads are inadequate for a two-dimensional stress analysis, a thickness analysis will be conducted to identify the effect of the bolt in restraining deflections at the joint. The thickness analysis will provide information on the stress concentrations, due to the application of a Q and an M near the bolt hole, by providing information on the stress field variations through the thickness.

B. TWO-DIMENSIONAL THICKNESS MODEL

The joint was also analyzed through the thickness in order to determine the thickness effect of a transverse shearing force and a bending moment on the plate when it is being held by some type of mechanical fastener. A plane stress membrane element provided by SAP IV [Ref. 6] was used in the arrangement as shown in Figure VI.B.1. Seventy-seven nodes and sixty elements were used to idealize a thin, 0.05 inches, section of the plate at the fastener. The moment was assumed to be created by a linearly varying tensile force along the thickness, where;

$$\sigma = \frac{M\,z}{I} \qquad \text{and} \qquad I = \frac{b\,h^3}{12}$$

The value of M is taken as unity for the analysis. The stress computed from above, times the effective area that

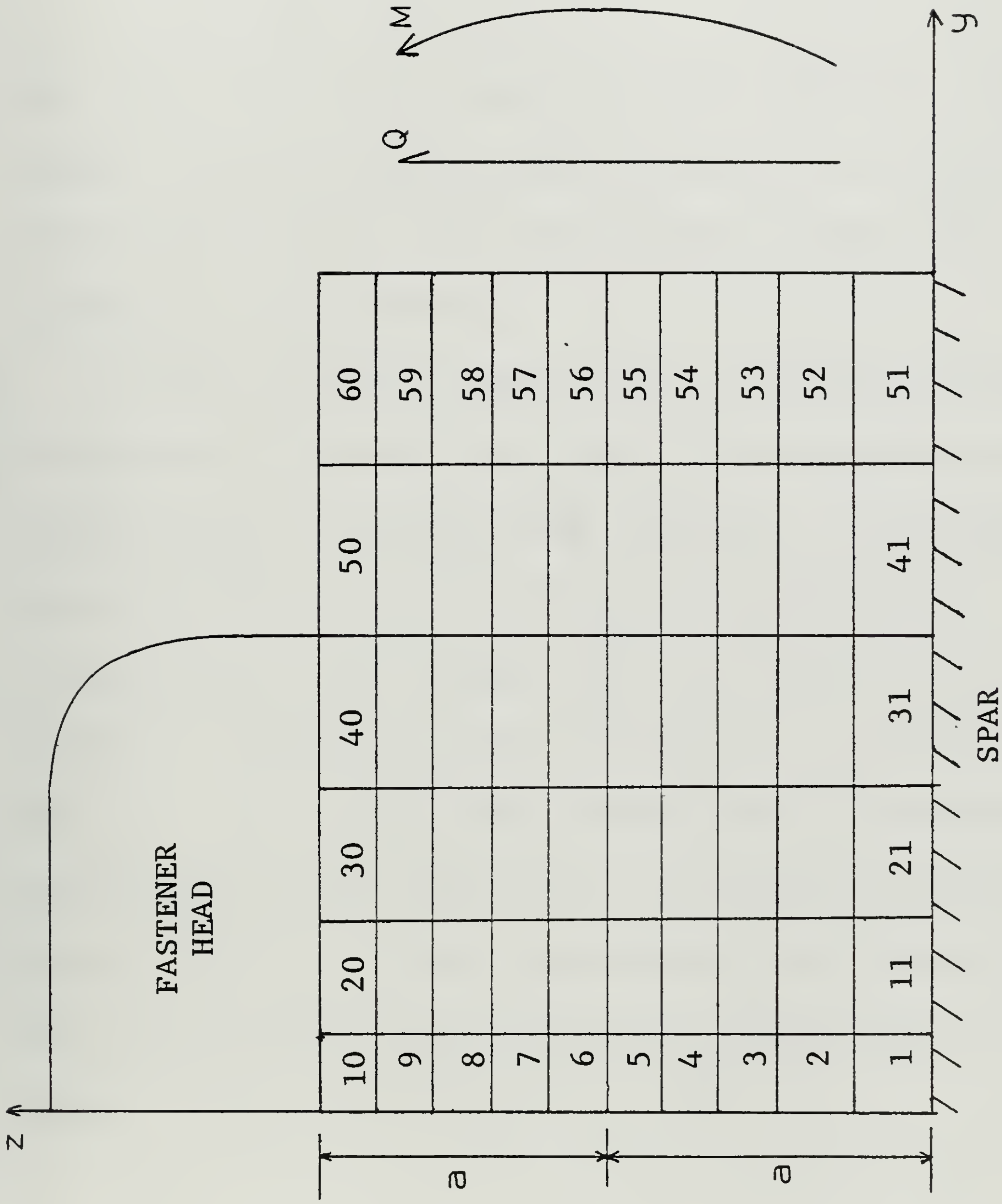

Figure VI.B.1 THICKNESS MODEL

it acts upon on the membrane, gives the force to be applied at each node. The thickness shearing forces were assumed to vary parabolically through the thickness, where;

$$\tau_{yz} = \frac{-Q}{2I} \left(a^2 - z^2\right)$$

The value of Q is also taken as a unit load/unit length. Again, by multiplying the above stresses by the effective area around a node, the forces acting on that node could be computed for the analysis.

As with the plate element model, defining the boundary conditions to represent the effect of the fastener on the plate is a job of finesse and intuition. It has been assumed that friction will prevent displacements in the y-direction at the fastener head, and that the plate can displace in both the y-direction and the z-direction at the spar. It has also been assumed that the fastener sufficiently pinches the plate as to prevent some displacement of the plate in the y-direction at the fastener body. The fastener head would also prevent displacement of the plate in the z-direction at the head.

Figures VI.B.2 and VI.B.3 show the stress distribution through the thickness and along the y-axis for a unit bending moment/unit length. The results indicate that the bending moment is not going to be the most critical load on the

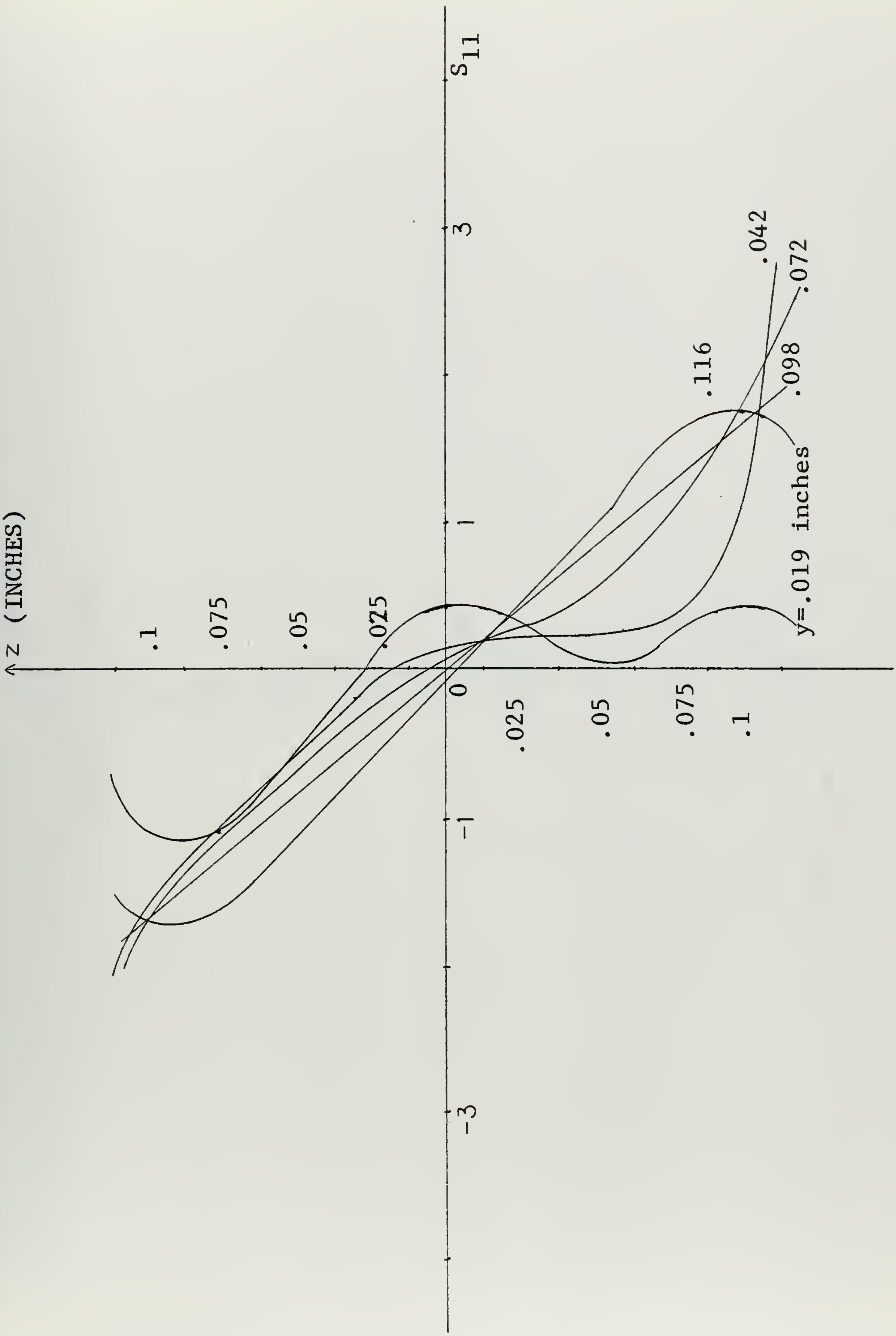

Figure VI.B.2 S_{11} VERSUS Z DUE TO UNIT BENDING MOMENT

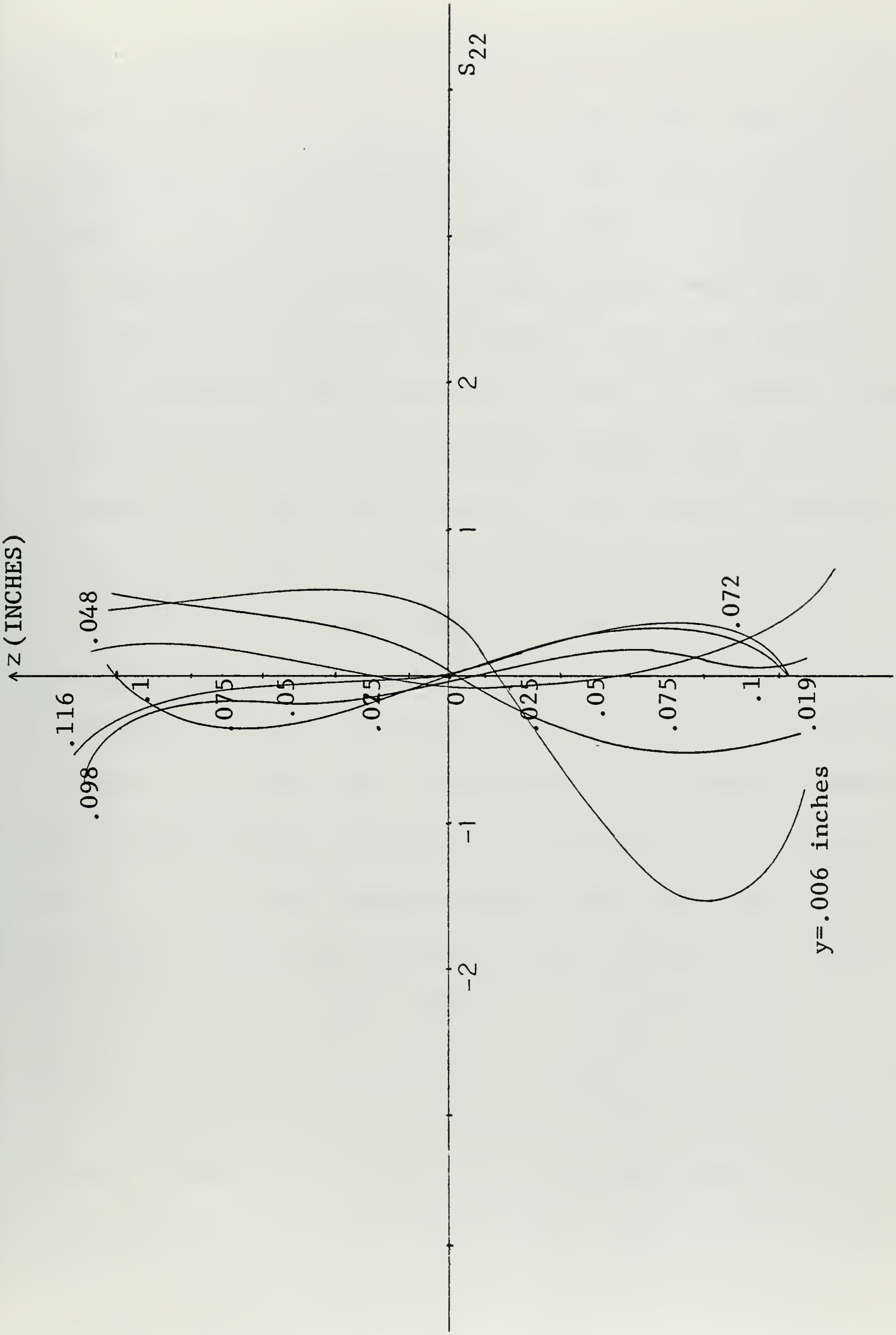

Figure VI.B.3 S_{22} VERSUS Z DUE TO UNIT BENDING MOMENT

joint. The maximum stress concentrations produced by the bending moment are only about Z, insignificant even in comparison with those produced by a unit tensile force/unit length on the plate. This maximum stress concentration is in compression at the edge of the fastener head, where the fastener can first pick up the load. One would expect these stresses in the z-direction, due to the bending moment, to be zero. Although the computer analysis does not show that they are zero, the values up to the fastener head are very close to zero. Once the fastener starts to restrict the deflections, though, there is a gradual increase in the stresses near the fastener body. The stresses in the y-direction, due to the bending moment, would be linear in a normal plate analysis. Since there is a fastener, though, the stress changes from what is essentially a linear distribution in z, to an odd distribution which is caused by the fastener re-straining the deflections of the plate. The majority of the load is picked up at the edge of the fastener head and then decreases towards the fastener body.

Figures VI.B.4, VI.B.5, and VI.B.6 show how the stress varies through the thickness and along the y-axis for the thickness shearing force/unit length. The results indicate that the thickness shear may be significant in the failure

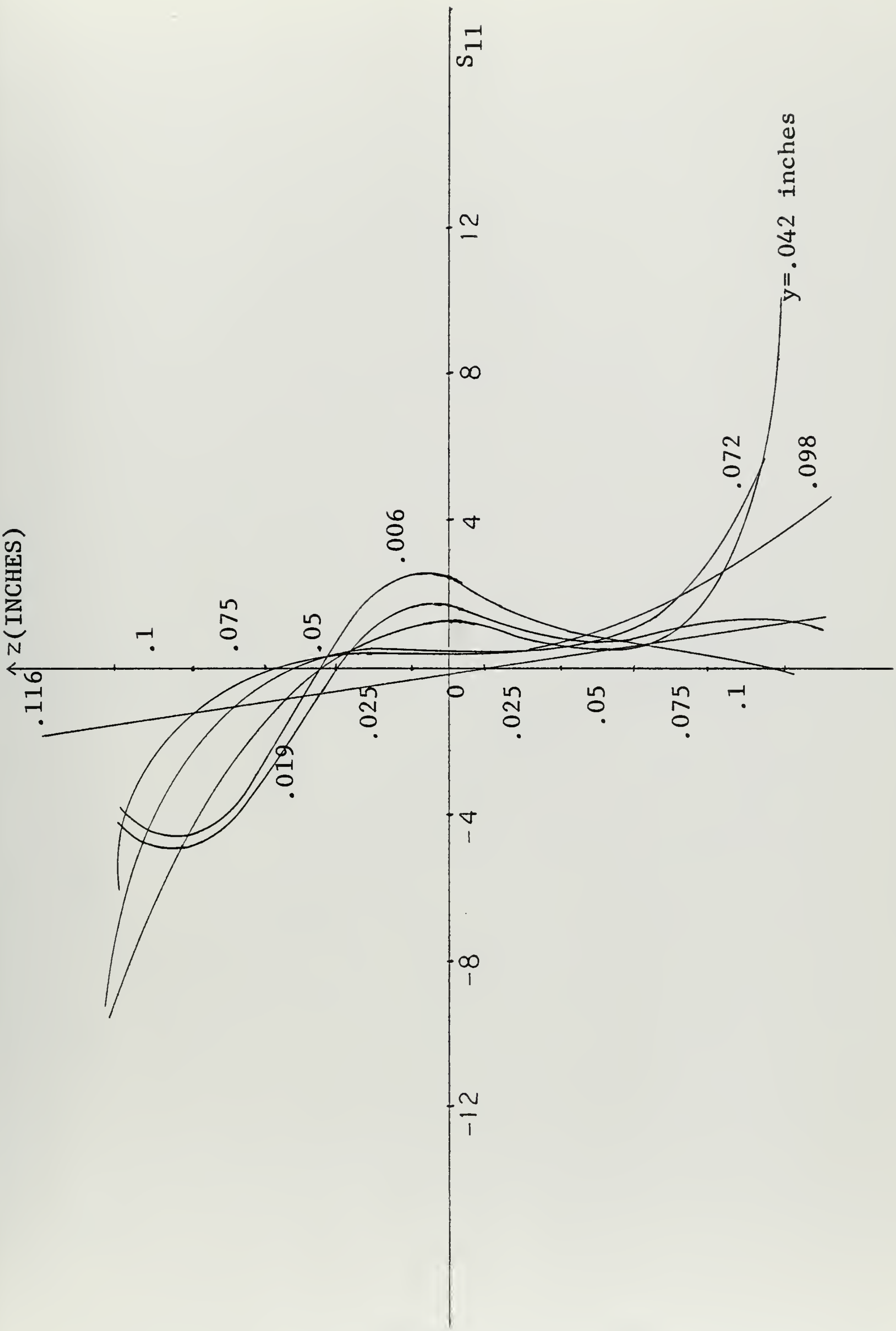

Figure VI.B.4 S_{11} VERSUS Z DUE TO UNIT THICKNESS SHEAR

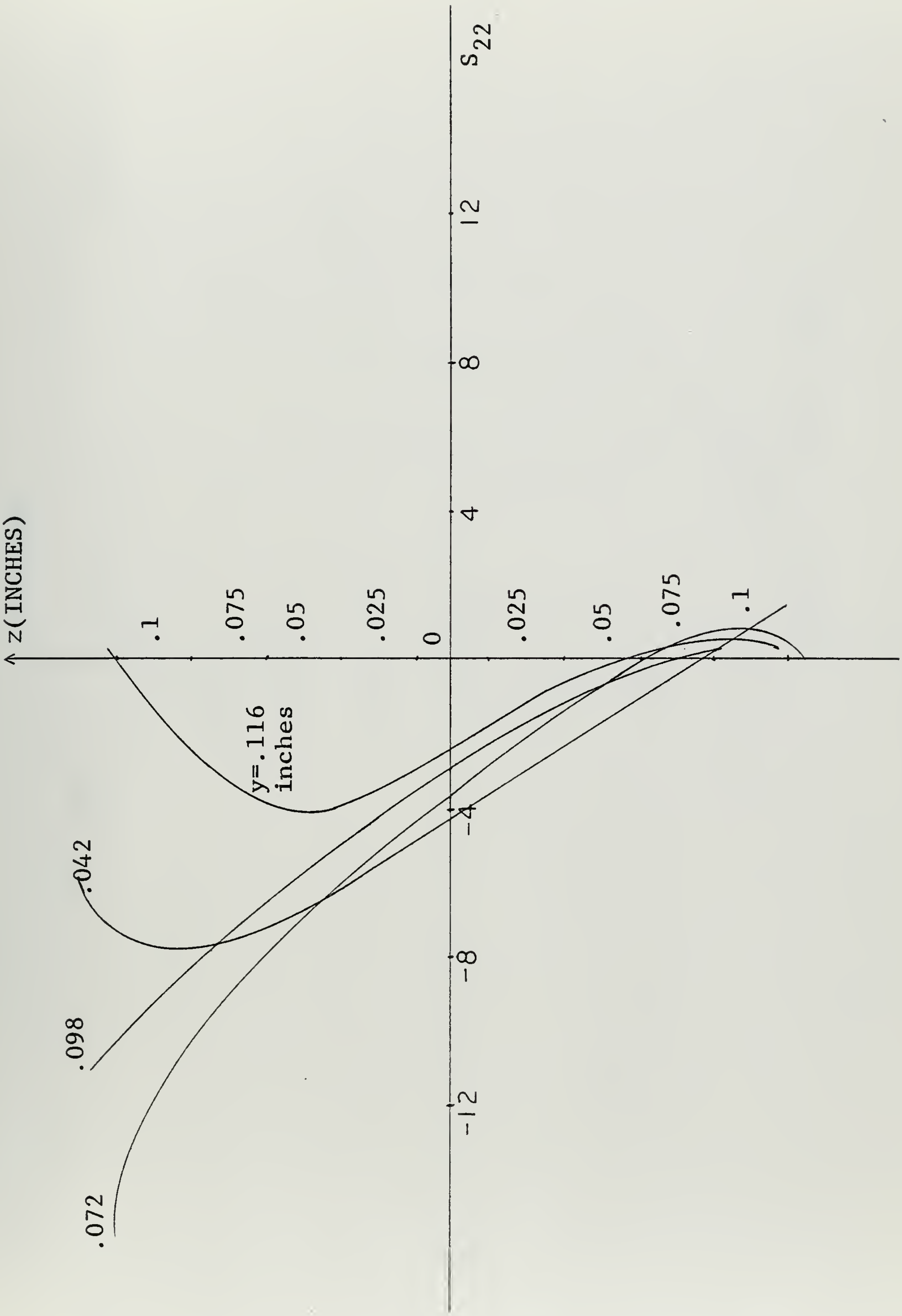

Figure VI.B.5 S_{22} VERSUS Z DUE TO UNIT THICKNESS SHEAR

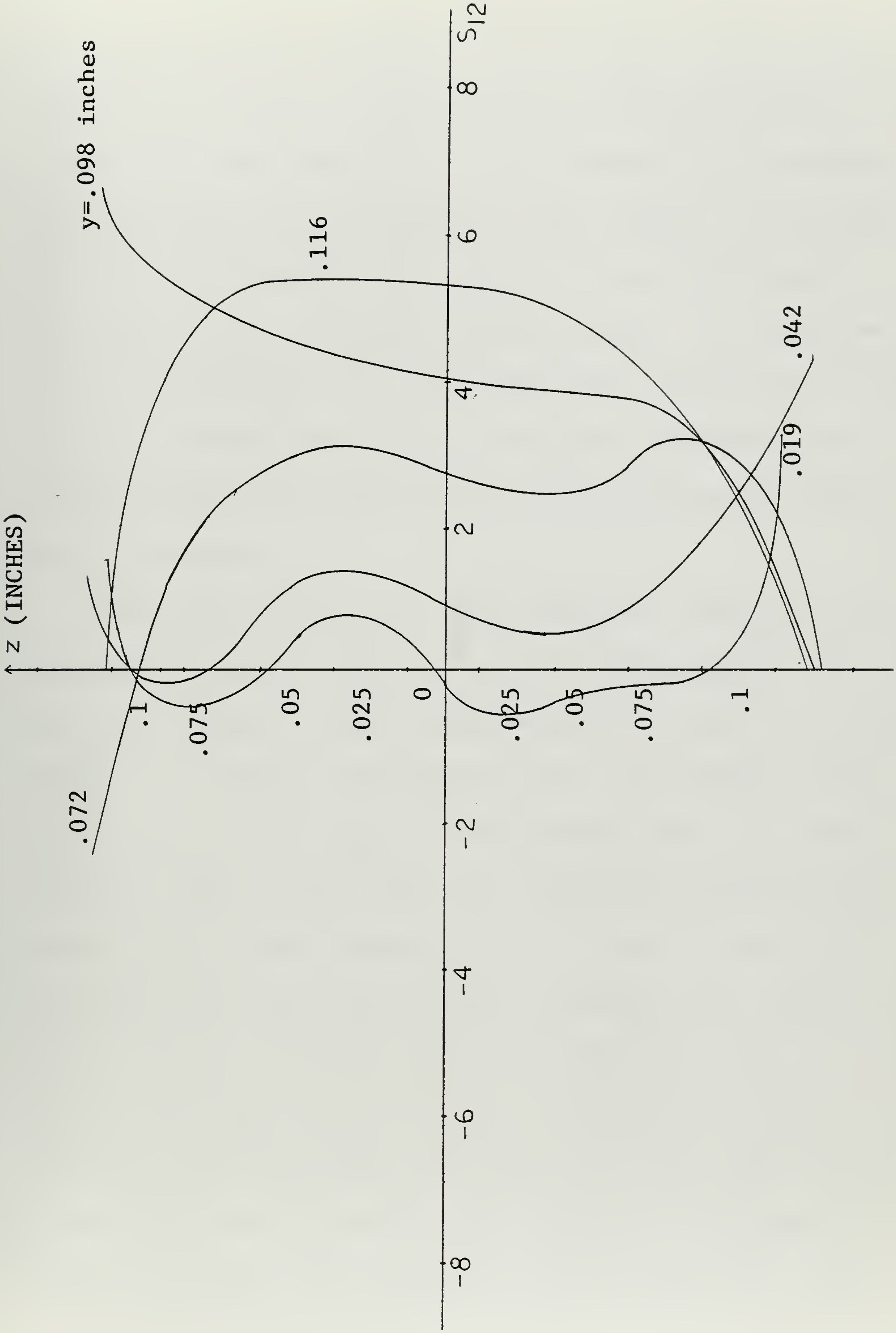

Figure VI.B.6 S_{12} VERSUS Z DUE TO UNIT THICKNESS SHEAR

of the joint due to hydraulic ram loading. As one would expect, the stress in the material in the z-direction is essentially zero where there is no restraint at the bottom of the plate, but it increases almost linearly to its maximum where the fastener head restricts the displacements and picks up the transverse shearing loads. The stresses in the y-direction due to the transverse shearing force start off in a linear distribution through the thickness, but rapidly change from linear due to the re-straining effect of the fastener.

The stress field due to the transverse shearing force is much the same as that due to the bending moment, but with much larger stress concentrations. As with the bending moment, the stress concentrations for the thickness shear are greatest at the edge of the fastener head. The results indicate that for the transverse shearing force/unit length, compressive stress concentrations of about 14 exist at the fastener head, and therefore the thickness shear may be the most critical load produced by hydraulic ram loading in the joint. Not only are skin joints not generally designed to consider large thickness shearing forces, but there are no readily available techniques to evaluate the thickness shear properties of a laminated fibrous composite. Most

epoxy matrices used in composites are such that they would crush under a large compressive thickness shear. The fibers do not have the strength to carry the high transverse shearing forces, either. Thus, the thickness shearing forces appear to be the most likely explanation of the failure of previously tested composite plates under hydraulic ram loading. Although the previous tests were done with clamped edges, the thickness analysis of the mechanically fastened joint is similar to that for a clamped edge. The Ultrasystems tests of hydraulic ram loading on aluminum plates were conducted with mechanically fastened joints, and therefore this analysis is probably a close representation of what probably happened around the joint. The thickness shear stress concentrations at the fasteners would explain the large scale unzipping that occurred during those tests.

TABLE VI.1

Material Properties

Material: Aluminum

Type: 6061-T6

E = 10,000,000 psi

G = 3,750,000 psi

ν = .33

W_t = .098 lb/cu in

Material: Graphite-Epoxy

Type: Narmco 5208/T 300

E_1 = 20,700,000 psi

E_2 = 1,470,000 psi

G_{12} = 750,000 psi

ν_{12} = .31

ν_{21} = .025

VII. FAILURE CRITERIA

The study of the fracture characteristics of fibrous laminated composites is a relatively new field of investigation that has a large number of inherent problems associated with it. Although the study of the fracture of a single lamina is fairly well established, it is still significantly affected by the fiber material used, such as graphite, boron, or others. The matrix or epoxy used, the presence of voids in the matrix, and many other factors also affect the failure characteristics.

Two main types of fracture exist for unidirectional composites, one is longitudinal splits parallel to the fibers, in which the failure starts in the matrix or at the fiber-matrix interface, [Ref. 3]. Since most matrices have very low strengths, only enough to transfer the loads and shears between the fibers, very low stresses in the material can cause this type of failure. Therefore, this is the mode of failure one would expect to occur first. The second type of fracture of the material is a transverse cracking of the lamina perpendicular to the fiber direction. This transverse failure requires considerably higher stresses

than those for the longitudinal fracture, since it requires the breaking of the fibers. Therefore, this type of failure is not as easily produced as is the longitudinal failure.

Both of the above fracture types can be explained by tensile forces in a lamina. If the tensile load is perpendicular to the direction of the fibers, then the lamina will fail longitudinally. If the tensile load is parallel to the direction of the fibers, then the lamina will fail transversely, breaking the fibers. A bending moment can also cause both types of fracture. Since a bending moment is just a linearly varying tensile force through the thickness, a bending moment applied perpendicular to the fiber direction will result in a longitudinal failure of the lamina. When the moment is applied parallel to the direction of the fibers, the lamina would then fail in the transverse mode. The thickness shear load, though, cannot be related to either of these two fracture modes. Very little has been done to explain fracture by Q, since it is not critical under normal aircraft flight loads.

The actual type of crack that occurs in composite laminates is usually a combination of both the longitudinal and the transverse cracking, since the laminates used in aerospace vehicles are generally not unidirectional, [Ref. 12].

A crack or fracture, such as the one along the clamped edge of a composite plate, shown in Figure II.4, is transverse in some layers, and longitudinal in others. For cracks and fractures in composites due to the hydraulic ram loading, where the plate failed at the attachments, the stresses in the material are dependent upon N, M, and Q, and the combination of the three makes the prediction of a particular failure mode very difficult.

Three failure theories are now being used to analyze the failure criteria of both unidirectional laminas and general laminates, [Ref. 7]. All three theories had their basis with isotropic, ductile, and homogeneous metals. The first two theories, maximum stress theory and maximum strain theory, although very easy to use, do not account for interaction between strengths, and are governed by the inequalities that determine the material's strength for a specific stress state. The third theory, distortional energy, has many variations that exist as the result of different basic assumptions. In general, though, the distortional energy theories have become the most widely accepted in predicting the strength limits of composite laminates, [Ref. 7]. These three theories and the yield surface they produce have been developed to the extent that

they can be used only for plane stress conditions. By specifying the stresses a bending moment and tensile force would produce on each individual lamina in the laminate, though, these theories may also be used to determine the yield surfaces applicable for M and N.

Although these theories might possibly be expanded to give an indication of the yield surfaces, including the transverse shearing force, Q, at the present time no such theory is available. Very little experimental work has been done to determine the material properties of any specific composite laminate, either. Therefore, there is essentially no information on the effect of transverse shear on the failure of composites.

VIII. JOINT DESIGN TO REDUCE HYDRAULIC RAM DAMAGE

Since it appears that a mechanically fastened joint in a fibrous laminated composite may fail under hydraulic ram loading, designs to prevent the joint from failing, or at least to restrict the damage, should be developed. One of the most critical areas of the joint is the channel groove, or similar cutout in the composite plate, that permits a smooth skin surface by keeping the fasteners flush with the skin. This cutout not only decreases the local thickness of the plate, and therefore its ability to carry loads, but it also increases the effect of a thickness shearing force and a bending moment on the joint. This cutout will create stress and moment concentrations, whose magnitudes are still unknown for the composite.

One way to reduce the effect of the groove in the composite is to not use it at all, and therefore not have a smooth skin. Although this would change the air flow over the skin because of boundary layer interruption, it would increase the ability of the joint to carry the loads created by hydraulic ram loading. It would also make the manufacturing procedure easier, less time consuming, and

less expensive. Cutting holes and channels in a composite plate not only alters the behavior and load carrying capability of the plate, but it is also a critical operation in which it is very easy to damage the plate.

Another method of decreasing the damage produced by hydraulic ram failure at the joint is to use a large diameter washer. This washer would allow better transfer of stresses to the fastener because of its larger contact area with the composite. If a smooth skin is desired, though, the washer would require even larger cutouts or channel grooves in the composite, and therefore might worsen the problem.

A third method of altering the joint design would be to use an expandable fastener. A fastener that could easily expand in length in order to allow better transverse shear stress and bending moment transfer to the adjacent skin sections and joints would be an exapndable fastener. This would probably be the most effective way to reduce or prevent damage to the joint, but it would also be the most difficult method to develop. No alterations to the joint size or type would probably be necessary, only a replacement of the fastener with one that is expandable. Developing such a fastener with mechanical techniques, or making use

of its material properties to make it expandable, would be a relatively new method of attachment and would require extensive study, testing, and money.

IX. CONCLUSIONS

The main objective of this study was to investigate the influence of a tensile force per unit length, N, a bending moment per unit length, M, and a transverse shearing force per unit length, Q, on a typical attachment used for aircraft fuel tank skins. These three types of loads are produced in the aircraft skin panel of a fuel tank due to the high fluid pressure on the skin caused by hydraulic ram loading. Although the finite element analysis of the loads around a mechanically fastened joint, using SAP IV [Ref. 6], was for an isotropic material instead of a composite material, it was concluded that the results for an isotropic material would give an indication of the relative magnitude of the stress concentrations around the joint, due to the loads produced by hydraulic ram loading. These results indicate that there is a need for closer examination of the response of composites to hydraulic ram loading.

Although the stress concentrations due to N on a mechanically fastened joint are high, it was found that they are less significant than those caused by the thickness shearing forces. Further, the bending moment, M, does not create stress concentrations that are as great as those

of either N or Q. Thus, although the failure at the fastener is due to a combination of N, M, and Q, the shearing force, Q, may be the major cause of the unzipping of aluminum plates from their mechanically fastened joints. It may also be the major cause of the edge failure of a clamped composite panel under hydraulic ram loading.

Although stress analysis of laminated fibrous composite joints was not undertaken, the results for an aluminum joint indicate that the hydraulic ram loading presents a formidable threat to composites used for the fuel tanks of military aircraft. Not only is little known of the thickness failure properties of laminated composites, but large scale damage in a material such as a graphite-epoxy composite is less acceptable than it is for aluminum, because of high material and repair costs.

To solve the problem of joint failure due to hydraulic ram loading, the type of attachments being used to join the aircraft skins to the stiffer members of a fuel tank may have to be changed or modified. The solution that would require less redesign of current aircraft designs is to develop an expandable fastener which would allow a transfer of thickness shearing loads to other attachments.

Continued testing of hydraulic ram loading phenomena is also necessary in order to better identify the many

factors that affect hydraulic ram loading in fuel tanks, such as fluid volume, stiffness of the entry wall, and others. Further testing is also desired to increase the available knowledge of failure modes, joint loading, and the response of different joining techniques to hydraulic ram loading. The use of a common fuel tank test cell would improve the ability to correlate data gained from hydraulic ram testing. Typical fuel tank test cells should be similar to those shown in Figures III.A.3 and III.B.1, in order to approximate the fuel tanks used in aircraft. One major problem exists with these test cells, though: they may not give an accurate representation of the total unzipping that might occur as a result of hydraulic ram loading. Although these test cells will give one an indication that unzipping has occurred if the unzipping is extensive, larger test cells would be needed to better understand the extent of unzipping that might occur.

X. RECOMMENDATIONS

1. Actual testing of the effect of hydraulic ram loading on an aircraft fuel tank joint, built of laminated fibrous composites, is needed to give a more accurate idea of the type of fracture to be expected. The testing should be done with mechanically fastened joints to investigate their effect on the failure of the composite.

2. Static testing of laminated composite samples to failure, for complete three-dimensional failure properties, would permit more accurate analysis of the composite under loads such as transverse shearing forces. Plane stress analysis of composites is insufficient to determine the failure at a joint.

3. A more in-depth study of joint response to hydraulic ram loading, using finite element techniques for three-dimensional problems, is also recommended. Two-dimensional techniques are insufficient to accurately explain the effects of layering the composite and the effects of transverse shearing forces and bending moments on the composite.

4. An investigation of stiffener and spar flexibility would also increase the understanding of load transferral

at the joint. This would provide additional knowledge to be used in limiting damage from hydraulic ram loading in fuel tanks.

5. Further investigations into joint design techniques to prevent hydraulic ram failure is also recommended. A better understanding of what is actually happening at a joint is therefore very necessary to improve on these techniques.

6. Testing of hydraulic ram loading is also necessary to determine the extent of unzipping that can be expected in composite skins. Since it is unknown whether or not a composite skin will unzip, this testing would not only be for determining how far down the spar or rib the skin will unzip, but whether it will unzip at all, and what kind of fracture is involved.

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